TRANSACTIONS

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

VOLUME 35

THIRTY-FIFTH ANNUAL MEETING
CHICAGO, ILL., JANUARY 28-31, 1929

SEMI-ANNUAL MEETING
LAKE-OF-BAYS, ONTARIO, CANADA, JUNE 26-28, 1929

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Officers and Council

AMERICAN SOCIETY *of* HEATING *and* VENTILATING ENGINEERS

1929

President .... THORNTON LEWIS
First Vice-President .... L. A. HARDING
Second Vice-President .... W. H. CARRIER
Treasurer .... W. E. GILLHAM
Secretary .... A. V. HUTCHINSON
Technical Secretary .... P. D. CLOSE

Council

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L. A. HARDING, *Vice-Chairman*

One Year	*Two Years*	*Three Years*
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JOHN HOWATT	ROSWELL FARNHAM	F. C. MCINTOSH
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Finance: W. T. Jones, *Chairman;* C. V. Haynes, F. C. McIntosh.
Membership: Roswell Farnham, *Chairman;* John Howatt, G. L. Larson.
Publication: A. C. Willard, *Chairman;* N. W. Downes, W. A. Rowe.

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COMMITTEES—1929

COMMITTEE ON RESEARCH

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S. R. Lewis, *Vice-Chairman*

F. C. Houghten, *Director*
O. P. Hood, *Ex-Officio Member*

One Year	*Two Years*	*Three Years*
Philip Drinker	A. R. Acheson	O. W. Armspach
S. R. Lewis	D. S. Boyden	R. S. Franklin
F. D. Mensing	R. V. Frost	F. E. Giesecke
W. A. Rowe	L. A. Harding	A. P. Kratz
A. C. Willard	F. B. Rowley	A. E. Stacey

Coördinated Research: J. C. Fitts, *Chairman* (*Nat'l Assn. H. & P. Contrs.*); R. M. Conner (*American Gas Association*).

Technical Advisory Committees

Temperature, Humidity and Air Motion: W. H. Carrier, *Chairman;* O. W. Armspach, S. C. Bloom, C. A. Bulkeley, Philip Drinker, W. A. Rowe and Perry West.

Infiltration: G. L. Larson, *Chairman;* A. W. King, A. P. Kratz, L. B. Lent and W. C. Randall.

Heat Transmission: A. P. Kratz, *Chairman;* D. R. Brewster, F. B. Rowley, C. G. Segeler and Perry West.

Pipe Sizes for Heating Systems: H. M. Hart, *Chairman;* S. E. Dibble, R. S. Franklin, F. E. Giesecke, C. V. Haynes and R. C. Morgan.

Heating and Ventilation in Its Relation to Health and Comfort: Philip Drinker, *Chairman;* John Aeberly, O. W. Armspach and C. P. Yaglou.

Radiation: R. A. Wolff, *Chairman;* C. W. Brabbée, R. V. Frost, J. D. Hoffman, C. H. B. Hotchkiss, J. F. McIntire, F. D. Mensing and A. C. Willard.

Garage Ventilation: E. K. Campbell, *Chairman;* W. H. Carrier, E. B. Langenberg and Thornton Lewis.

Air Cleaning Devices: F. B. Rowley, *Chairman;* H. E. Birkholz, Albert Buenger, E. Vernon Hill and H. C. Murphy.

COMMITTEES—1929

Atmospheric Dust and Smoke: E. B. Langenberg, *Chairman;* E. Vernon Hill, S. R. Lewis and H. C. Murphy.

Oil Burning Devices: L. E. Seeley, *Chairman;* James Breese, Jr., G. S. Meikle and H. L. Tapp.

Effect of Wind and Weather Conditions on the Heating Loads: R. S. Franklin, *Chairman;* W. L. Fleisher, J. F. Hale, E. B. Langenberg, S. R. Lewis, F. R. Still and A. C. Willard.

Testing and Rating Unit Heaters: D. E. French, *Chairman;* O. K. Dyer, G. E. Otis, H. W. Page, W. A. Rowe, J. H. Shrock and L. C. Soule.

SPECIAL COMMITTEES

Advisory Committee on International Heating and Ventilating Exposition: H. P. Gant, *Chairman;* E. B. Langenberg, J. I. Lyle, J. F. McIntire, F. R. Still, H. L. Whitelaw, E. K. Webster, H. C. Murphy, A. S. Armagnac, D. S. Boyden, A. C. Edgar, C. V. Haynes, W. H. Carrier and Roswell Farnham.

Committee on Meetings Program: H. H. Angus, *Chairman;* A. J. Nesbitt and F. B. Rowley.

Committee on Code for Testing Unit Heaters: D. E. French, *Chairman;* L. C. Soule, W. A. Rowe and H. W. Page; representing *Industrial Unit Heater Association*: G. E. Otis, *Chairman;* O. K. Dyer and J. H. Shrock.

Committee for Interpreting Code for Rating Low-Pressure Heating Boilers: L. A. Harding, *Chairman;* R. V. Frost and F. C. Houghten.

Committee to Prepare Code for Testing and Rating Concealed Radiators: G. E. Otis, *Chairman;* W. H. Carrier, R. N. Trane, R. C. Malvin and F. C. Houghten.

Guide Publication Committee: S. R. Lewis, *Chairman;* W. H. Carrier, C. V. Haynes and J. F. McIntire.

Nominating Committee: F. D. Mensing, *Chairman.*

Cleveland Chapter—H. M. Nobis
Illinois Chapter—John F. Hale
Kansas City Chapter—N. W. Downes
Massachusetts Chapter—J. F. Tuttle
Michigan Chapter—J. H. Walker
Minnesota Chapter—Glenn C. Morgan
New York Chapter—H. B. Hedges
Western New York Chapter—O. K. Dyer
Ontario Chapter—A. S. Leitch
Pacific Northwest Chapter—E. O. Eastwood
Philadelphia Chapter—R. C. Bolsinger
Pittsburgh Chapter—T. M. Dugan
St. Louis Chapter—C. A. Pickett
Wisconsin Chapter—G. L. Larson

Officers of Local Chapters

1929

Cleveland
Headquarters, Cleveland
Meets: Second Friday in Month
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Secretary, R. G. DAVIS
Treasurer, H. M. NOBIS
Board of Governors: C. F. EVELETH, F. A. WEAGER, D. E. HUMPHREY

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Meets: Second Monday in Month
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Treasurer, AUGUST KEHM
Board of Governors: J. H. O'BRIEN, R. B. HAYWARD, E. P. HECKEL

Kansas City
Headquarters, Kansas City, Mo.
Meets: First Monday in Month
President, C. C. CLEGG
Vice-President, E. K. CAMPBELL
Secretary, F. A. KITCHEN
Treasurer, C. W. PLASS
Board of Governors: H. C. HENRICI, HENRY NOTTBERG, BEN C. NAYLOR

Massachusetts
Headquarters, Boston
Meets: First Monday in Month
President, T. F. MCCOY
Secretary, J. S. WEBB
Treasurer, L. J. MCMURRER
Board of Governors: T. F. MCCOY, G. H. GLEASON, PHILIP DRINKER, J. W. BRINTON, DAVID MOULTON, L. R. STETSON, E. C. WHITTAKER

Michigan
Headquarters, Detroit
Meets: First Monday after 10th of Month
President, W. G. BOALES
Vice-President, E. E. DUBRY
Treasurer, R. K. MILWARD
Secretary, E. H. CLARK
Board of Governors: L. L. MCCONACHIE, W. J. WHELAN, H. M. STEPHEN

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Headquarters, Minneapolis
Meets: Second Monday in Month
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Vice-President, D. M. FORFAR
Secretary-Treasurer, M. S. WUNDERLICH
Board of Governors: A. J. HUCH, W. F. UHL

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Headquarters, New York
Meets: Third Monday in Month
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Secretary, E. B. JOHNSON
Treasurer, F. E. W. BEEBE
Board of Governors: R. A. WOLFF, C. W. BRABBÉE, A. J. OFFNER

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Headquarters, Buffalo
Meets: First Monday in Month
President, O. K. DYER
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Second Vice-President, JOSEPH DAVIS
Treasurer, C. H. LOVE
Secretary, D. J. MAHONEY
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Headquarters, Toronto, Can.
Meets: First Monday in Month
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Vice-President, H. J. CHURCH
Secretary-Treasurer, J. S. PATERSON
Board of Governors: A. J. DICKEY, R. W. MCHENRY, E. B. SHEFFIELD

Pacific Northwest
Headquarters, Seattle, Wash.
Meets: Second Thursday in Month
President, E. L. WEBER
Vice-President, E. O. EASTWOOD
Secretary, M. ANDERSON
Treasurer, W. W. COX
Board of Governors: W. L. DUDLEY, C. F. TWIST, W. E. BEGGS

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Headquarters, Philadelphia
Meets: Second Thursday in Month
President, A. C. EDGAR
Vice-President, H. G. BLACK
Secretary, L. C. DAVIDSON
Treasurer, E. N. SANBERN
Board of Governors: R. V. FROST, A. J. NESBITT, LEE NUSBAUM

Pittsburgh
Headquarters, Pittsburgh
Meets: First Monday in Month
President, H. LEE MOORE
Vice-President, E. W. STITT
Secretary, W. W. TEAGUE
Treasurer, PETER O'NEILL
Board of Governors: F. A. GUNTHER, R. H. HEILMAN, M. M. TIMMERMAN

St. Louis
Headquarters, St. Louis
Meets: First Wednesday in Month
President, F. J. MCMORRAN
First Vice-President, E. A. WHITE
Second Vice-President, C. G. BUDER
Secretary, R. M. ROSEBROUGH
Treasurer, E. H. QUENTIN
Board of Governors: L. WALTER MOON, A. E. HUMPHREYS, J. I. DENEILLE, R. E. GRAVES

Wisconsin
Headquarters, Milwaukee
Meets: Third Monday in Month
President, W. F. NOLL
Vice-President, F. G. WEIMER
Secretary, V. A. BERGHOEFER
Treasurer, MARTIN ERICKSON
Board of Governors: H. M. MILLER, P. C. DOWNEY, G. L. LARSON

TRANSACTIONS

of

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

No. 819

THIRTY-FIFTH ANNUAL MEETING, 1929

FIVE hundred and twenty-four people registered at the 35th Annual Meeting of the Society held in Chicago, Ill., at the Edgewater Beach Hotel, Jan. 28-31, 1929, with Pres. A. C. Willard presiding. The record of accomplishments may be briefly summarized as follows: (1) approval of revisions to the Code for Testing Low-Pressure Heating Boilers; (2) adoption of a Code for Rating Low-Pressure Heating Boilers burning solid fuel, so that manufacturers will have a uniform method to use and in that way provide the user with a yardstick to compare boilers; (3) approval of a standard for heat transmission symbols, definitions, etc.; (4) voting on several amendments to the Constitution and By-Laws; (5) election of two honorary members, and, (6) endorsing the Council's actions for making the Society's monthly JOURNAL a part of the new publication, *Heating, Piping and Air Conditioning,* commencing in May, 1929.

Oliver J. Prentice, representing the Chicago Association of Commerce, made the address of welcome and President Willard gave a fitting response.

The Chairman of the Board of Tellers of Election, E. N. McDonnell, made the following report:

Report of Tellers of Election

For President:	Thornton Lewis	540
For Vice-President:	L. A. Harding	550
For Second Vice-President:	W. H. Carrier	546
For Treasurer:	W. E. Gillham	548
For Members of the Council (for 3 years):	E. B. Langenberg	549
	G. L. Larson	549
	F. C. McIntosh	549
	W. A. Rowe	548

Members of Committee on Research (for 3 years):	O. W. Armspach	562
	R. S. Franklin	560
	F. E. Giesecke	561
	A. P. Kratz	562
	A. E. Stacey	561

Scattering votes were recorded for various other members.

TELLERS OF ELECTION
E. N. McDONNELL, *Chairman*
M. T. CLOW
WM. G. BOALES
O. W. ARMSPACH

The Secretary, A. V. Hutchinson, gave the Report of the Council and the Secretary's Report:

Report of Council

The organization meeting of the Council was held in New York, January 26, 1928 with Pres. A. C. Willard presiding and the personnel of Council Committees was announced. Arrangements were made for a Technical Advisor of the Society and contract for this service was made with Perry West, Newark, N. J.

A Committee to revise the Constitution in respect to membership requirements was appointed in March and a Constitution for the Pacific Northwest Chapter was approved, and changes in the Constitution of the Philadelphia Chapter were okayed. A Charter was granted to Indiana members for the formation of an Indiana Chapter.

An invitation to participate in the World Engineering Congress from E. A. Sperry, of New York, was favorably acted upon and a Committee headed by W. H. Carrier was requested to prepare a paper for presentation in Tokyo, October, 1929.

S. R. Lewis, of Chicago, was designated by the Council as the Society's member on the *National Research Council* for a period of three years and the Society also took membership in the *National Fire Protection Association.* Additional space at the headquarters office of the Society was obtained and necessary equipment installed in June. The Council nominated Messrs. Armspach, Chicago; Franklin, Boston; Giesecke, Austin, Texas; Kratz, Urbana, Ill.; Stacey, Newark, N. J.; for membership on the Research Committee prior to October 1 as required by the regulations.

The preparation of the Code of Minimum Requirements for printing was authorized and this will come to the members in looseleaf form for convenient use, each binder to have the name of the member imprinted on the cover.

The Council selected West Baden, Ind., as the meeting place for the Semi-Annual Meeting, June, 1928, and chose C. R. Ammerman, as Chairman of the Committee on Arrangements. The result of this most successful meeting was the formation of the Indiana Chapter, and the Council believes that the splendid work done by the group of men in handling the summer meeting forecasts the success of the new chapter.

The invitation of the Illinois Chapter for the Annual Meeting, 1929, was accepted and the offer of the Ontario Chapter to handle the Summer Meeting, 1929, was heartily approved. The Annual Meeting for January, 1930 is to be held in Philadelphia at the same time as the International Heating and Ventilating Exposition, which is to be held at the Commercial Museum. The Council now has before it, invitations to Massachusetts for the Summer of 1930 and a very urgent invitation from the Cleveland Chapter for the winter January, 1931.

The Council felt that the technical affairs of the Society were increasing to such an extent that an assistant secretary trained specially in heating and ventilating engineering should be added to the staff and this decision resulted in the employment of P. D. Close of Chicago for the year 1929.

A very important decision made by the Council was in connection with the publication of THE JOURNAL as a separate part of the new magazine, *Heating, Piping and Air Conditioning,* to be published in Chicago. The details of this transaction were sent to members in a letter under date of January 10, in which it was pointed out that each member would receive the new magazine beginning in May, and which also outlined the value of this arrangement, both in service to the Society and manufacturer in the in-

dustry, and the financial benefit that would accrue to the organization. The contract was signed on January 10, 1929 and it is to be effective for three years and can be renewed for two more.

The usual formal actions were taken by the Council in regard to resignations, dropping members for non-payment of dues, reinstating members who desire to reaffiliate. The Council has endeavored to carry out the wishes of the members as expressed through the men representing the various districts of the country and has acted at all times with the idea in mind of safeguarding the interests of the membership.

Respectfully submitted,

The Council.

Report of Secretary

The activities of the Society have expanded greatly this year and include the technical phases, research and publication work. All of this is reflected in the specific reports of the finance, membership, research, publication and the technical committees.

All Society programs have been handled by a Council Committee during the past year under the able leadership of Prof. F. B. Rowley, and as a result the last two meetings of the Society have had record attendances.

Two chapters have been organized, one in Seattle, Wash., and the other in Indianapolis, Ind. During the year, President Willard has visited nearly every Chapter and has kept the members closely in contact with all Society projects and policies.

Among the outstanding accomplishments on record for 1928, are *first,* the addition to the headquarters staff of a Technical Secretary, P. D. Close, of Chicago, who entered upon his duties in New York, January 14, 1929; *second,* the arrangement made to place THE JOURNAL in the new magazine, *Heating, Piping and Air Conditioning,* published in Chicago; *third,* the completion and publication of the Code of Minimum Requirements soon to be mailed to members and *fourth,* the sponsorship of the International Heating and Ventilating Exposition to be held in Philadelphia at the same time as the 36th Annual Meeting in 1930.

In addition to this the Committee on Research has been very active and has entered into several more cooperative agreements with colleges, the Laboratory in Pittsburgh has produced reports of far-reaching importance, and the Society has received recognition from many other engineering bodies and public bodies.

One of the important invitations accepted by the Society was to participate in the World Engineering Congress in Japan, 1929, and a Committee appointed by the Council is preparing a paper for presentation.

Several technical committees of the Society have been very active with the result that a Revised Code for Testing Heating Boilers and a Code for Rating Low-Pressure Steam Heating Boilers for Solid Fuel are to be presented at this meeting. A Code for Testing Unit Heaters is being prepared and reports will be made on the status of the Code for Garage Heating and Ventilating and a Code for Air Cleaning Devices.

The Heat Transmission Committee of the National Research Council has requested approval of its work upon definitions, symbols and units, etc., for heat transmission work.

Several Society reports are working under the procedure of the *American Standards Association* on the many engineering projects which are being sponsored.

The Finance Committee had the books of the Society audited and found that the budget has been most successful this year and the Society's finances and reserves are in splendid condition. The response of members to its request for early payment of dues was met very promptly and was of great assistance in the demonstration of Society affairs.

In the matter of the membership, 200 new names were added to the Society's roster and under the direction of John F. Hale of Chicago, the interest of all members was requested in increasing the membership. One of the serious problems of the future is maintenance of the membership at its increasing strength. During the year two past presidents have been lost by death as well as several charter members.

The prospect for 1929 holds forth the possibility of a greater influence by the

Society in the profession of engineering, an opportunity for more practical service to the membership and at the same time bring greater aid to the layman in providing a scientific solution for his problems in heating, ventilating, comfort and health.

Respectfully submitted,

A. V. HUTCHINSON, *Secretary.*

W. T. Jones, chairman of the Finance Committee gave the Report of the Certified Public Accountant as follows:

Report of Certified Public Accountant

January 15, 1929.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
29 WEST 39TH STREET
NEW YORK CITY

Gentlemen:

I have completed the examination of the books of account and records of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York City, for the year ended December 31, 1928, and submit herewith the following exhibits and comments:

EXHIBIT

"A" BALANCE SHEET—DECEMBER 31, 1928
Schedule
No. 1—Marketable Securities

"B" STATEMENT OF INCOME AND EXPENSES OF THE SOCIETY—FOR THE YEAR ENDED DECEMBER 31, 1928
Schedule
No. 4—Salaries

"C" STATEMENT OF INCOME AND EXPENSES OF THE PUBLICATIONS—FOR THE YEAR ENDED DECEMBER 31, 1928
Schedule
No. 2—Cost of JOURNAL
No. 3—Cost of GUIDE
No. 4—Salaries

"D" COMPARISON OF BUDGET—SOCIETY ACTIVITIES

"E" COMPARISON OF BUDGET—PUBLICATIONS

CASH

A verification was made of the Cash on Deposit by direct communication with the banks and reconcilement of the amounts reported to me with the balances shown by the books of the Society.

MARKETABLE SECURITIES

The Securities kept with the Bankers Trust Company for safe-keeping were verified by direct communication. A schedule appended hereto of the Securities owned by the Society on December 31, 1928, discloses that the cost exceeded the market value by $757.78.

ACCOUNTS RECEIVABLE

There is shown below classified as to years a summary of the Dues Receivable from members. A detailed check was made of the cash received against the serially numbered Dues Ledger Cards and all dues were accounted for:

1928 Unpaid Dues	$ 8,711.00
1927 Unpaid Dues	3,752.77
1926 Unpaid Dues	819.76
1925 Unpaid Dues	174.50
1924 Unpaid Dues	20.00
TOTAL UNPAID DUES	13,478.03
LESS: Prepaid Dues	301.41
DUES RECEIVABLE	$13,176.62

The Dues Receivable were reviewed with the management for the purpose of determining the collectability thereof and as a result the sum of $6,000.00 was added to the Reserve carried on the books to cover probable losses during realization. No change was made to the Reserve of $1,000.00 already provided to cover other Accounts Receivable as it is considered sufficient.

A summary of the members active on December 31, 1928, follows:

Members	1462
Associate Members	396
Juniors	123
Students	1
Honorary	1
	1983

INVENTORIES

The following Inventory of TRANSACTIONS taken on December 31, 1928, was submitted for my verification:

Year	Volume	Number	Price	Amount
1921	27	216	1.00	$216.00
1922	28	164	1.18⅖	194.18
1923	29	132	1.97½	260.70
1924	30	262	1.38¼	362.21
1925	31	289	0.96⅙	275.01
1926	32	140	1.26⅘	177.52
				$1,485.62

TRANSACTIONS covering the years 1927 and 1928, Volumes 32 and 33, respectively, have not been published as yet, therefore, a Reserve in the sum of $5,000.00 has been provided to cover the cost of producing both volumes.

All other Inventories and Deferred Charges were determined either by computation or count.

ACCOUNTS PAYABLE

Amounts due Trade Creditors were determined by trial balance of the unpaid invoices on file. Of the total Accounts Payable, aggregating $9,826.67, the sum of $9,542.00 covers GUIDE and YEAR BOOK costs.

Of the dues charged to Senior and Associate Members 40 per cent has been reserved for the Research Laboratory in accordance with *Section* 5, Article 3, of the By-Laws. The sum payable to the Research Laboratory as and when the Dues Receivable will have been realized in cash is $11,783.38. In addition there is due the Research Laboratory the sum of $6,227.20, representing that portion of the profit resulting from the GUIDE, which has been computed as follows:

INCOME FROM GUIDE	$37,550.22	
COST OF GUIDE	25,385.30	
GROSS PROFIT		$12,164.92
OVERHEAD EXPENSE:		
Amount charged to GUIDE Calendar Year 1927	1,944.93	
ADD: One-half of Increase of 1928 Expenses over 1927	277.21	2,222.14
		9,942.78
DEDUCT: Chapter Meeting Expenses		1,000.00
		8,942.78
Bonus to Publication Management and Employees (59%)		2,715.58
NET PROFIT FROM THE GUIDE FOR RESEARCH LABORATORY		$6,227.20

ACCRUED ACCOUNTS

Bonus to the publication management, the Clerical Staff and the Solicitor computed in accordance with resolutions approved by the Council during the current year has been included as part of the year 1928 operations.

GENERAL FUND

An analysis of the General Fund showing a reduction of $318.59 for the past year follows. Had it not been for the fact that the amount expended on the Code of Minimum Requirements exceeded the Budget Provision by $2,584.52, the General Fund would have shown an increment.

GENERAL FUND—DECEMBER 31, 1927—PER FORMER REPORT		$35,798.54
DEDUCT: Loss on Society Operations for the year ended December 31, 1928 from Statement of Income and Expenses	$4,114.12	
ADD: Profit from Publications for the year ended December 31, 1928, from Statement of Income and Expenses	3,732.53	
NET LOSS FOR THE YEAR		381.59
GENERAL FUND—DECEMBER 31, 1928—PER BALANCE SHEET		$35,416.95

Respectfully submitted,

FRANK G. TUSA,
Certified Public Accountant.

BALANCE SHEET

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, NEW YORK CITY

December 31, 1928

ASSETS

CASH				
On Deposit— General Fund		$3,739.59		
On Deposit—Research Fund		1,323.89		
			$5,063.48	
On hand			100.00	
				$ 5,163.48
MARKETABLE SECURITIES—PER SCHEDULE				
General Fund		25,596.53		
Add: Accrued Interest		300.84		
			25,897.37	
Research Fund			3,045.61	
				28,942.98
ACCOUNTS RECEIVABLE				
Dues		13,176.62		
Less: Reserve for Doubtful		8,970.00		
			4,206.62	
Advertisements	$35,059.36			
Guides	499.53			
Transactions	108.00			
		35,666.89		
Less: Reserve for Doubtful		1,000.00	34,666.89	
				38,873.51
INVENTORIES				
Transactions			1,485.62	
Journal Paper			208.75	
Emblems			98.91	
Picture Frames			82.50	
				1,875.78
LIBRARY				300.00

Furniture and Fixtures	$4,799.97	
Less: Reserve for Depreciation	2,710.80	$2,089.17
Deferred Charges		
Publications		
Journal Wrappers and Mailing	221.76	
Society		
Printing and Stationery	359.69	581.45
		$77,826.37

LIABILITIES

Accounts Payable		$ 9,826.67
Due Research Laboratory		
Dues	$11,783.38	
1928 Guide Profit	6,227.20	18,010.58
Accrued Accounts		
Bonus—Publication Management	3,640.60	
Bonus—Employees	728.12	
Bonus—Solicitor	233.95	
Professional Services	600.00	5,202.67
Reserve for Transactions		
1927	2,500.00	
1928	2,500.00	5,000.00
Funds		
General	35,416.95	
Research	4,369.50	39,786.45
		$77,826.37

NOTE "A." There were no contingent liabilities reported to me and as far as could be ascertained none existed.

NOTE "B." This Balance Sheet is subject to the comments contained in the letter attached to and forming a part of this report.

EXHIBIT "A"

Budget Comparison—Society Activities

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

For the Year Ended December 31, 1928

	Actual 1927	Actual 1928	Budget Provision	Increases Decreases
INCOME				
Dues—1928	$24,288.07	$24,738.49	$25,000.00	$ 261.51
Initiation Fees	2,465.15	2,499.00	3,000.00	501.00
Sale of Emblems and Certificate Frames	122.50	237.50	100.00	137.50
Interest Earned	1,347.54	1,175.14	1,000.00	175.14
Profit from Sale of Securities	161.70	200.00		200.00
TOTALS	$28,324.96	$28,850.13	$29,100.00	$ 249.87
EXPENSES				
Salary—Secretary	$2,500.00	$2,750.00	$2,750.00	
Salary—Clerical	4,372.51	5,420.00	5,000.00	420.00
Rent—Room No. 602	1,000.00	1,000.00	1,000.00	
Professional Services	360.00	360.00	400.00	40.00
Postage	1,153.94	1,219.84	1,500.00	280.16
General Printing	853.52	970.36	1,000.00	29.64
Yearbook	743.43	862.78	800.00	62.78
Cost of Emblems and Certificate Frames	52.33	113.31	100.00	13.31
Traveling—Secretary	953.23	1,335.90	1,000.00	335.90
Traveling—President	1,500.00	687.51	1,000.00	312.49
Meetings—Annual and Semi-Annual	3,018.02	3,099.58	3,000.00	99.58
Local Chapter Meeting Allowance	1,094.90	1,000.00	1,000.00	
Council Meetings		358.17	500.00	141.83
Exhibit—Power Show	83.65	64.35	100.00	35.65
$2.00 per Member for Journal	3,908.00	3,966.00	4,000.00	34.00
$1.00 per Member for Transactions	1.954.00	1,983.00	2,000.00	17.00
Dues		60.00		60.00
Publicity Expense	618.39	195.73		195.73
Special Boiler Code Committee	628.62	568.41	500.00	68.41
APPORTIONABLE EXPENSES—60%				
Rent—Room No. 603	1,236.20	1,577.85	1,260.00	317.85
Office Expense	1,180.76	1,385.46	1,300.00	85.46
Office Supplies	364.07	430.52	500.00	69.48
Allowance for Depreciation of Furniture and Fixtures	243.37	271.80	300.00	28.20
	$27,818.94	$29,680.57	$29,010.00	$ 670.57
PRIOR YEAR'S DUES REALIZED	$ 4,006.15	$ 5,072.55	$ 6,000.00	$ 927.45
CODE OF MINIMUM REQUIREMENTS	$ 1,000.00	$ 3,584.52	$ 1,000.00	$2,584.52

EXHIBIT "D"

Budget Comparison—Publications

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

For the Year Ended December 31, 1928

	Actual 1927	Actual 1928	Budget Provision 1928	Increases Decreases
INCOME				
JOURNALS				
Advertising	$20,190.28	$21,802.15	$21,500.00	$ 302.15
Sales of Journals	999.63	1,087.38	1,000.00	87.38
$2.00 per Member from Annual Dues	3,908.00	3,966.00	4,000.00	34.00
	$25,097.91	$26,855.53	$26,500.00	$ 355.53
TRANSACTIONS				
$1.00 per Member from Annual Dues	$ 1,954.00	$ 1,983.00	$ 2,000.00	$ 17.00
Copy Sales	672.50	259.61	500.00	240.39
	$ 2,626.50	$ 2,242.61	$ 2,500.00	$ 257.39
GUIDE				
Advertising	$27,460.22	$31,539.94	$30,000.00	$1,539.94
Sale of Guides	3,997.49	6,010.28	4,500.00	1,510.28
	$31,457.71	$37,550.22	$34,500.00	$3,050.22
SALE OF REPRINTS AND BOOKS	$ 1,171.80	$ 846.48	$ 1,200.00	$ 353.52
	$ 1,171.80	$ 846.48	$ 1,200.00	$ 353.52
TOTALS	$60,353.92	$67,494.84	$64,700.00	$2,794.84
COSTS AND EXPENSES				
COST OF PUBLICATIONS				
Journal	$12,689.70	$14,297.93	$14,920.00	$ 622.07
Guide	21,794.53	25,385.30	24,200.00	1,185.30
Transactions	2,442.03	2,620.31	2,500.00	120.31
Reprints	796.55	722.66	1,000.00	277.34
	$37,722.81	$43,026.20	$42,620.00	$ 406.20
EXPENSES				
Salary	$ 2,500.00	$ 2,750.00	$ 2,750.00	
Salary—Clerical	3,511.00	4,074.00	4,500.00	426.00
Professional Services	240.00	240.00	300.00	60.00
Postage	354.58	337.77	400.00	62.23
Traveling	749.91	475.46	500.00	24.54
APPORTIONABLE EXPENSES—40%				
Rent—Room No. 603	864.00	864.00	840.00	24.00
Office Expense	726.93	860.83	800.00	60.83
Office Supplies	243.15	122.98	400.00	277.02
Allowance for Depreciation of Furniture and Fixtures	162.25	181.20	200.00	18.80
	$ 9,351.82	$ 9,906.24	$10,690.00	$ 783.76
	$47,074.63	$52,932.44	$53,310.00	$ 377.56

EXHIBIT "E"

The Report of the Committee on Research was presented by the Chairman, Samuel R. Lewis:

Report of Committee on Research

The Laboratory has continued in accordance with the general policy of encouraging University cooperation laid down by this administration.

Support by our friends in the industry has been generous, and the custom of asking for five-year pledges has been discarded in favor of one which asks for annual promises with an unwritten understanding that these continue indefinitely during our good behavior.

I am delighted to announce that under this arrangement the necessary funds to keep us going during 1929, up to at least our present scale, have been promised during the last two months of 1928.

We have not encroached on the reserve fund. The endowment fund grows slowly but is at least established and if, having retired to private life after these years of the white glare of publicity, I can now get rich in my private business, I might slip the endowment fund something in my will.

I do seriously remind the Chapters of perhaps forgotten resolutions regarding this endowment fund. Some of them, I believe, will at this time, as soon as I quiet down, wish to be heard from on the subject.

The child of the Laboratory, the Association for Coordinating Thermal Research, is organized and functioning and has been financed.

Our technical activities at the various laboratories are evidenced by the various papers presented at this meeting.

Our latest cooperative agreements with Purdue and Harvard Universities are well under way, but it is too early to expect very elaborate reports.

We now have these arrangements with ten educational institutions besides the Bureau of Mines and not counting our Weather Bureau Cooperation.

The latest agreement, just consummated, is with Yale University, and will especially cover oil burner research.

This Weather Bureau cooperation is one of the most interesting activities of the Laboratory, as it has grown to include also cooperation with the health departments of 13 large cities. We may prove, before we get through, undreamed of things concerning the effect on health of dust and smoke. We seem to be headed for investigations into violet rays, smoke prevention, dust filters, hay fever cures and many other angles under the Technical Advisory Committee on Atmospheric Dust and Smoke. I had one letter alleging that it was popularly believed that we were going to select the most healthy city in the United States, and this Chamber of Commerce wanted to help us select the fair city of its own nativity.

There are no formal reports from any of the technical advisory committees, but every one of them has functioned effectively in 1928 and I have had letters from every chairman, which indicate that every chairman knows all about what has been going on in his line.

It is a pleasure to know that partly through impetus from the Research Laboratory we now have more papers for presentation at our meetings than we know what to do with.

The Laboratory is now holding back, permitting careful study and checking and digestion, a number of valuable papers from its staff, while our publication committee no longer has to beg for papers, but can, on the other hand, pick and choose and turn away with high disdain.

The chairmen of our technical advisory committees are at this meeting. Practically every learned professor who will speak at any of our sessions belongs to the staff of our Research Laboratory.

Financially the situation is as follows:

Income for 1928

Received from member dues		$12,722.96
Received from interest on deposits		144.38
Received from contributions		12,690.32
Promised from dues		4,275.00
Promised from GUIDE		6,200.00
Overdue 1928 contributions		2,625.00
		38,657.66
Outgo for 1928		32,867.07
To surplus		$ 5,790.59
Our budget was	$35,175.00	
We expended	32,867.07	
Beating the budget	$ 2,307.93	

We have written promises of contributions for 1929 amounting to $9,600.00 made prior to January 1, 1929, with many of our best friends yet to hear from, so that the outcome financially is more favorable by far than it has ever been before.

It is with real regret that I relinquish the fun of being Chairman of the Committee on Research.

It is with real pleasure, however, that following the traditions of the office I have led the way in nominating and electing my distinguished successor as chairman, Prof. L. A. Harding.

This man has given the Society very generous service on the Committee on Research, also as Chairman of the Committee on the Code of Minimum Requirements, and more recently on the Council.

He combines astuteness as a scientist with commercial ability of a high order.

I am glad that I shall still have one year on the Committee on Research under his leadership.

Respectfully submitted,

S. R. LEWIS, *Chairman.*

As a preliminary to the discussion of the subject of boiler testing and rating, President Willard stated that for a number of years the Society's Code for Testing Low-Pressure Heating Boilers had been in effect but that progress had made revisions necessary and from time to time this work had been undertaken by special committees. In the work of the Committee on Rating of Boilers, it was found that the Testing Code needed to be brought into agreement with testing practices of today and that the Council had placed this matter in the hands of Mr. Kellogg's Committee and he was glad to introduce Mr. Kellogg, who would present the report of the Committee on Code Testing for Low-Pressure Heating Boilers, which had been studied in detail by the Council and the Society, and unanimously approved for presentation to the Society for action.

Alfred Kellogg of Boston then presented the recommendations of his Committee on the Revision of the Society's Code for Testing Low-Pressure Heating Boilers and recommended its approval by the membership. This motion was seconded by Dean F. Paul Anderson, who stated that it was his pleasure to second the motion for adoption of this Code; *first,* because of the necessity for a Code of this kind and the opportunity it gives us to present to the engineering world, facts in reference to the performance of boilers; *second,* because, when it needs modification, the Society has provided the necessary machinery in the form of a competent committee to safeguard the users.

President Willard invited discussion on this subject and in the absence of any further suggestions, a vote on adopting the revised Boiler Testing Code was taken and the motion unanimously carried.

Mr. Kellogg proceeded with the presentation of the report of his Committee on Rating of Low-Pressure Heating Boilers.

He reviewed the work undertaken by the Committee on Code for Rating Low-Pressure Heating Boilers and F. C. Houghten, director of the Research Laboratory demonstrated by means of black-board sketches, the method of using boiler performance charts in obtaining an A.S.H.&V.E. rating. Mr. Kellogg then stated that the Committee submits the Code to the Council and Society for consideration and adoption.

President Willard called for a discussion of the subject and stated that the Council was unanimous in its endorsement of the Code for Rating, and L. A. Harding moved that the Code dated January, 1929 be adopted as one of the Society's Standard Codes.

A substitute motion was offered by Dr. Brabbée suggesting that the Code be presented to the boiler manufacturers for further discussion. The motion was seconded by Homer Linn and on vote, was defeated.

The original motion offered by Mr. Harding, seconded by Mr. Carrier, on vote, was passed.

It was pointed out by Professor Willard that further revisions of the Society's Code would be needed in the future and it was the opinion of the officers that provisions should be made for the future handling of this work so that suggestions received from the membership and others who might use the Code, could be classified and incorporated as occasion might require. Therefore, it seemed justifiable to advise the Society that President-Elect Lewis and the present incumbent have conferred on the matter and agreed that a committee should be appointed for the purpose of interpreting, continuing and extending the Testing Code and the logical men were L. A. Harding, the new Chairman of the Committee on Research, R. V. Frost, a man of well-known experience in boiler testing work, and F. C. Houghten, Director of the Society's Research Laboratory.

COMMITTEE ON RATING LOW-PRESSURE HEATING BOILERS

REPORT OF JANUARY, 1929

THE June, 1928, report of your committee which was submitted to the Society at the Semi-Annual meeting was referred back to the committee for reconsideration, but without any definite recommendations. Your committee, therefore, considered it advisable that the Resolution adopted by the Society should be referred to the Council. This was done, accompanied by the request that the status of the committee be reconsidered and, if necessary, that it be reinstructed.

The recommendations of the Council were passed at its meeting on October 1st and were printed in the October issue of the JOURNAL, page 769. The committee was instructed to prepare a code which would cover the determination of the output without covering Selection, the preparation of the Code for Selection being allotted to the Committee on Code of Minimum Requirements. The Council stated "That the Boiler Rating Code requested is to embody the best judgment of the Committee, in the light of West Baden and other discussions, of what will constitute the most workable Code."

Your committee therefore prepared a new code complying with these instructions, and submitted it to the Council, which approved it at their meeting on December 6th. The new code is appended to this report.

In its previous reports your Committee has recommended that the Rating outputs be based on safety from priming and on the temperature of the flue gases. The reports have shown that such a method is fundamentally sound and is logical, and that the numerical value adopted for the temperature of the flue gases was, to a large extent, immaterial, but that the meaning that would be associated with Rating values so determined would depend on the numerical values adopted. The reports have pointed out that the information conveyed by such a rating output was only part of the data required in comparing boilers and that the other operating characteristics of stack draft and rate of burning must be given consideration. The reports have also shown that if a standard method of rating were adopted, then the Code for Selection could fix the relationship between the maximum demand load and the Rating output and could vary the ratio between these with the boiler size, or could change it from time to time to meet service conditions.

Previous reports have suggested that the determination of the Rating output could be made to include some measure or limitation of the fuel bed operation, such as rate of burning, and the Committee has suggested that this was desirable, but the suggestion did not receive any active support. In spite of this, a study of the discussion at the Semi-Annual meeting indicates that, without knowing how it was to be accomplished, the speakers were in favor of an arbitrary method of rating by which the numerical value would express a combined measure of the heating surface and grate area. The value of these two factors is expressed in terms of operating characteristics, under controlled conditions, by the flue-gas temperature and rate of burning per unit area, respectively.

The Code which is now submitted adopts this principle. It retains the priming limitation and establishes an arbitrary output equality between boilers of different ratios of heating surface and grate area by balancing the one against the other. Such a method is logical as a means of expressing an average numerical value and, in effect, corresponds to that used for rating warm air furnaces.

The Code as submitted assumes that a series of tests have been made and a Performance Chart plotted; for the purpose of this Code a complete chart is not necessary, and the tests could be restricted to the range of outputs in which that of the Rating value will probably occur.

The Council believed that the Code should also include the determination of the Rating output of hot-water boilers which are tested as such. This has not been included in the Code submitted, but its arrangement and titling are such that hot-water boilers can be included in the future. Such ratings would be similar to those for low pressure steam except that the priming limitation would be replaced by one covering freedom from water hammer, and probably some measure of the hydraulic resistance of the boiler. The Committee did not consider it advisable to fix such limitations because the Society has no code for testing hot-water boilers.

The Council instructed the Committee to revise the 1925 issue of the Code for Testing Low-Pressure Steam Heating Boilers to be in agreement with the Rating Code. The Rating Code as submitted does not require any changes in the principles of the Testing Code. The Committee is submitting a revision, in which the 1925 issue is made more complete by the addition of fuller instructions and advice. There also was careful consideration of the various conditions under which boilers would be tested. It was recognized that the more frequent application of the Code is in the testing of boilers on a test block when there would be fuller facilities. In addition to this, however, a code issued by the Society should be suitable for general use.

The Committee recognized that there has been a demand for a shorter code, in fact a committee was appointed a few years ago to draw up one, but nothing was accomplished. The revised code takes care of this in the revisions of the forms. The 1925 issue calls for complete measurements of the boiler and other details which would only be required under very special circumstances. It is, therefore, logical that these can be omitted and not included in the standard forms, leaving it to the engineer to include them in his special report. It was also recognized that boilers are often tested without attempting to obtain a complete heat balance, and without having a full analysis of the fuel. These conditions have been taken care of and the tabulated items have been reduced to the smallest number consistent with the various uses to which the Code may be put.

The 1925 issue has always carried a foot-note that forms could be obtained at a nominal cost. It was evidently the original intention that the forms called for by the Code should be printed so that they could be used by the testers and for reports. The Committee does not submit the revised Code as a series of form sheets which could be duplicated, but its divisions are such that such forms could be drawn up and printed should there be a demand for them, and should the Society decide to supply them.

Respectfully submitted by the Committee,

ALFRED KELLOGG, *Chairman*
F. C. HOUGHTEN P. NICHOLLS
SAMUEL R. LEWIS L. E. SEELEY

Code for the Rating of Heating Boilers Burning Solid Fuel

(*Issue of January, 1929*)

PURPOSE

THE purpose of giving heating boilers Rating output values as defined by this Code is to establish a method of determining from the operating characteristics a Rating which will give reasonably fair comparative values for boilers of different sizes or design.

This Code is not intended to supplant the more correct engineering practice of determining the available output of boilers for specified operating conditions by a study or analysis of the complete data given by "performance charts;" nor does it preclude the assigning of other rating output values to meet purchasing specifications or a specified set of operating conditions. The rating outputs as determined by this Code are intended for average conditions and for the use of purchasers not competent or desirous of making comparisons and selection from performance charts.

RATING DESIGNATION

Rating outputs conforming to this Code shall be known as A.S.H.&V.E. Rating.

This Code fixes the maximum output that may be designated as the A.S.H.&V.E. Rating. The output allowed by this Code may require a higher draft or other operating characteristic than the manufacturer would desire when listing the boiler for average use; this Code allows that a lower rating output may be chosen and may be listed as the A.S.H.&V.E. Rating.

TEST SPECIFICATION

Low-Pressure Steam Heating Boilers.

The tests of the boiler shall have conformed to the Code for Testing Low-Pressure Steam Heating Boilers of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

A series of tests shall have been made and a Performance Chart plotted showing—for the purpose of this Code—flue-gas temperature, rate of burning fuel of a calorific value of 12,500 and draft at smokehood against rate of output.

The average carbon dioxide (CO_2) in the flue gases within the range of the Rating output defined by this Code shall not be less than 12 per cent by volume by flue-gas analysis when burning anthracite or coke, nor less than 10 per cent when burning bituminous coal.

RATING DEFINITION

LOW-PRESSURE STEAM HEATING BOILER

The Rating Output of a low-pressure steam heating boiler burning solid fuel shall be determined as follows:

Hand-Fired Boilers.

From the Performance Chart and the curves of Fig. 1 of this Code, determine—

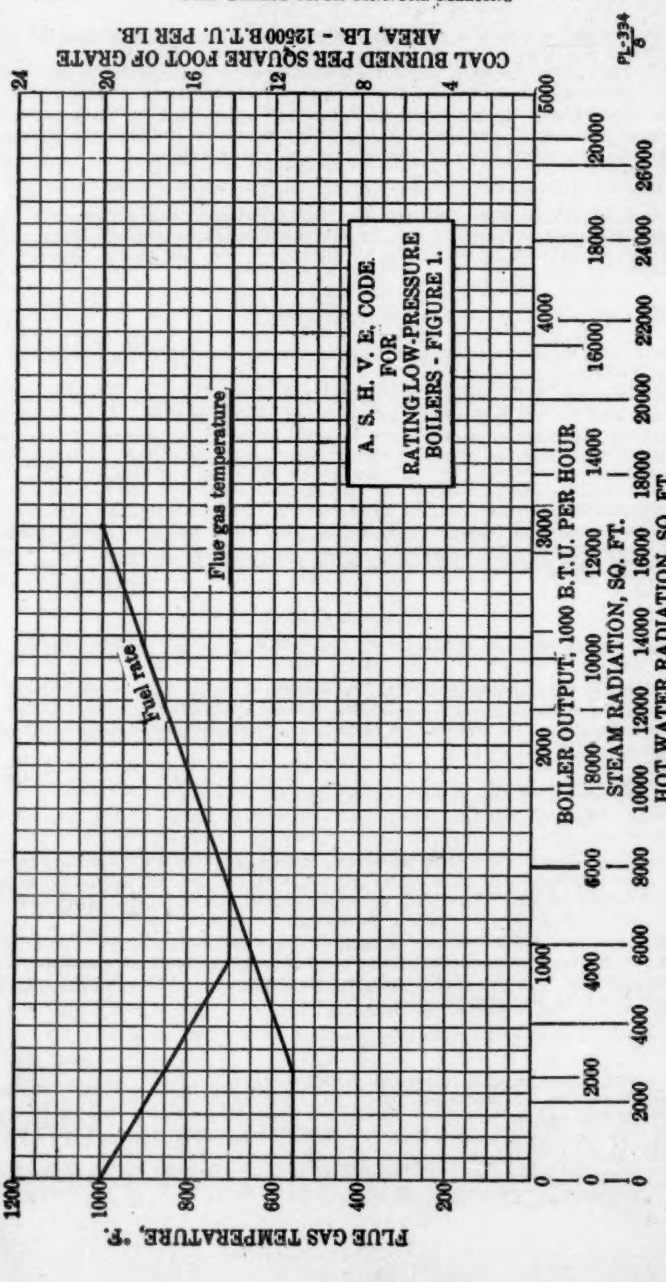

FIG. 1. CHART FOR DETERMINING A.S.H.&V.E. RATING OF LOW-PRESSURE HEATING BOILERS

(*a*) The rate of output at which the temperature of the flue gas leaving the boiler was that fixed by the flue-gas temperature curve of Fig. 1.
(*b*) The rate of output at which the average rate of burning per square foot of nominal grate area was that fixed by the fuel rate curve of Fig. 1.

The A.S.H.&V.E. Rating output for hand-fired boilers shall be the lower of the two outputs "*a*" and "*b*" plus 30 per cent of their difference, provided the priming at the output so determined does not exceed two per cent moisture.

Mechanical-Fired Boilers.

The A.S.H.&V.E. Rating output for mechanical-fired boilers that have been tested as a combined unit shall be that fixed by the flue-gas temperature curve of Fig. 1, provided the priming at the output so determined does not exceed two per cent moisture.

STANDARD RATINGS

There shall be two standard ratings, to be termed the Anthracite Rating and the Bituminous Rating, respectively.

Manufacturers shall determine and publish both of the standard ratings for boilers which are sold for use with either anthracite or bituminous coal. It is recognized, however, that there are types which are designed primarily for use with a particular fuel, and therefore that fuel would be used in determining the ratings. Ratings determined with other than the two standard fuels shall be published as *SPECIAL RATINGS*, and the fuel used in the test shall be specified by type, bed or other manner necessary clearly to define it. The same limiting conditions and test specifications shall be used for special rating tests as for the standard, except that when using fuels high in free hydrogen, or when using low-grade fuels, it shall be permissible to lower the allowable average CO_2 content of the flue gases to a minimum of 8 per cent.

STANDARD FUELS

The following descriptions specify the standard fuels as they apply to the purpose of this Code:

Anthracite. Any anthracite which by proximate analysis has not more than 8 per cent volatile on a moisture and ash-free basis. There shall be no limitation to the size of fuel permissible, but the size must be stated. It is recommended, however, that stove size (2½ to 1 9/16 in.) be used.

Proximate chemical analyses usually include the moisture and ash. The percentage volatile on a moisture and ash-free basis is computed by simple proportion and is:

$$\text{Actual volatile percentage} \times \frac{100}{100 - (\text{moisture} + \text{ash})\ \text{percentage}}$$

Bituminous Coal. The coal to be used in bituminous tests may be caking or non-caking (free burning) and shall contain not less than 35 per cent volatile on a moisture- and ash-free basis. There shall be no limitations to size of fuel permissible, but the size used, and whether it is caking or non-caking, must be stated.

(The practices in marketing bituminous coals are not standardized. Caking coals are but rarely available to definite sizes; sized non-caking coals are more available. It is recommended that as sized bituminous coals become available a size of 3 by 2 in. be standardized for test purposes.)

RATING DATA

Essential data required in statement of Rating are:
(*a*) Fuel, kind and size used
(*b*) Rating output
(*c*) Average draft at smokehood in inches of water, at Rating output
(*d*) Average flue-gas temperature at Rating output
(*e*) Average combustion rate, pounds of coal of 12,500 Btu per square foot of (equivalent) grate area per hour.

DISCUSSION

ALFRED KELLOGG: It would, perhaps, prove interesting to many of you if we should rehearse (in part at least) the history of the many attempts by this Society and other organizations to formulate a satisfactory code for rating low-pressure heating boilers. As you well know, all such efforts for one reason or another came to naught, and the time available at this meeting precludes going into this phase of the subject so familiar to many of you.

The instructions given this Committee at the time of its appointment, were that a standard code for *rating* low-pressure heating boilers should be drawn for the guidance of boiler manufacturers, in determining output; and that the data so obtained should be clearly stated in the manufacturers' catalogs, to be known as the A. S. H. & V. E. RATING. This would not preclude manufacturers from publishing output based on other conditions of operation, but the statement of the Society's rating would provide a yardstick by which purchasers could compare one boiler with another, regardless of make or design. This work was undertaken at the request of the U. S. Dept. of Commerce.

This Committee, very early in its work, concluded that it would be possible to afford the boiler-buying public what might be called a *ONE FIGURE RATING*. That, briefly explained, is this: that from performance curves obtained by the manufacturer from actual tests, definite points thereon could be charted or published in simplified form so that the purchaser of a boiler could know that the boiler would carry him through any extreme weather condition he would be likely to encounter.

The question of efficiency under such conditions is considered of secondary importance, because of the fact that a one figure rating would naturally be one that would carry it through the peak loads which might occur two or three times during the winter, probably never longer than a week at any one time, and under such conditions the owner could well afford to forget all about efficiency for the time being. That is what we all do.

Starting with that premise, we next endeavored to obtain the advice and experience of both the individual manufacturer and of the three organizations that had put in much time in trying to solve the problem, and we therefore requested these organizations, including all boiler manufacturers, to cooperate with us. In most cases our overtures met with a gratifying response.

Some manufacturers, however, did not seem to understand our objective, and as a result many meetings have been held during the past two years. One important hearing covering three days was held at Pittsburgh in April last, attended by about 30 representatives of the manufacturers and several contractors, and great interest was taken in the work of the Committee at those meetings. Our Committee has received valuable suggestions from various sources, some of which we have embodied in the several reports that have been placed before you from time to time.

Keep in mind, however, the objective of the work on which we are engaged: that is, to establish, if possible, the simplest rating factors that can be utilized by *NINETY PER CENT OR MORE* of the purchasing public.

As a result of the hearing at Pittsburgh, we submitted the report presented to the Society at the West Baden meeting. One or two members of the Com-

mittee were unfortunately unable to attend the meeting and the Committee has felt that perhaps its position was not clearly understood at the time. However, there were certain suggestions made by those present, some of which have been embodied in the report that you have before you.

In the present report there is a chart which we are going to explain. In the curve of flue-gas temperature, the temperature line rises sharply at about the 1000 sq ft rating figures shown at the bottom of the chart.

It was realized that 90 per cent of the total boiler sales are of 1200 sq ft or less capacity, therefore it was felt that in those small sizes it would be permissible to allow a temporary increase in flue-gas temperature above normal, keeping in mind the very short period that boilers have to operate at maximum output. Under normal conditions the gas-temperature conditions would be much lower, accompanied also by a satisfactory degree of operating efficiency.

There are, however, other features of the chart that should be clearly brought out, and I am going to ask Mr. Houghten, who worked upon that feature, to further explain the chart reproduced on the blackboard.

F. C. HOUGHTEN: How to rate a boiler in accordance with the rating code which is before you is the question that I am required to answer. In order to do so the two charts, Figs. 2 and 3, have been prepared.

Fig. 2 is a performance chart for a boiler. It is a picture of what the boiler will do under different operating conditions. Let us now consider it the performance chart of a particular boiler that we want to rate in accordance with the A. S. H. & V. E. rating code.

You are probably acquainted with such performance charts. We plot on the horizontal, or the rate of heat output axis, the various items which enter into the performance of a boiler. For instance, the flue-gas temperature is plotted against output, giving curve *A*, showing that the flue-gas temperature rises from 380 to 970 F as the boiler output increases from 1000 to 7000 sq ft equivalent direct radiation.

There is also plotted the rate of combustion, in pounds of fuel burned per

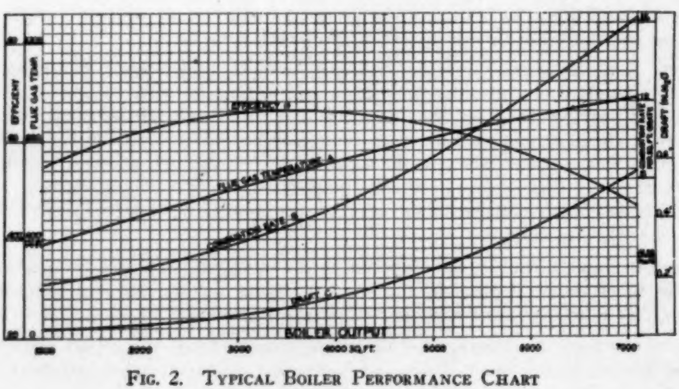

FIG. 2. TYPICAL BOILER PERFORMANCE CHART

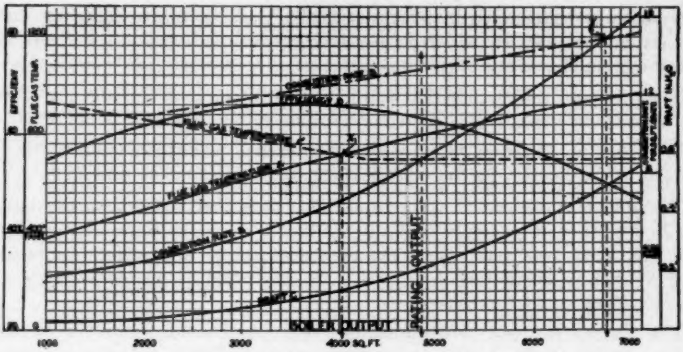

FIG. 3. METHOD OF DETERMINING A.S.H.&V.E. BOILER RATING

hour per square feet of grate, against the output, giving curve *B*. Likewise, the draft required and the efficiency resulting are given by curves *C* and *D*.

Now, how do we rate this boiler? In the chart, Fig. 1 of the rating code, the rate of combustion and the flue-gas temperature limitation curves are plotted against boiler output. To rate the boiler which is represented by the performance chart, Fig. 2, you may take the limitation curves from Fig. 1 of the rating code and superimpose them upon the performance chart, or you may take the flue-gas temperature curve and the rate of combustion curve from the performance chart and superimpose them on Fig. 1 of the rating code.

The chart, Fig. 3, is the same performance chart of the boiler to be rated as was shown in Fig. 2. In addition to the four solid line performance curves given in Fig. 2, the flue-gas temperature limitation curve and the rate of combustion limitation curve from Fig. 1 of the rating code are superimposed on this performance chart giving the broken line curves *A′* and *B′*.

To find the rating of the boiler find the intersection *X* of the flue-gas temperature performance curve *A* and the flue-gas temperature limitation curve *A′*. Also the intersection *Y* of the rate of combustion, performance curve *B* and the rate of combustion limitation curve *B′*.

X and *Y* give boiler outputs of 4030 and 6740, respectively. Provided the performance curves were plotted from tests showing the required percentage of CO_2 and not over 2 per cent of moisture in the steam, the rating of the boiler will be the lower of the two outputs *X* and *Y* plus 30 per cent of their difference, or it is $4030 + 0.30\ (6740 - 4030) = 4843$.

According to the code the A. S. H. & V. E. rating of this boiler is stated as,

RATING DATA

(*a*) Fuel .... stove size anthracite
(*b*) Rating output .... 4843 sq ft
(*c*) Smokehood draft at rating output .... 0.21 in. water
(*d*) Average flue-gas temperature at rating output .... 800 F
(*e*) Average rate of combustion at rating output .... 8.7 lb per sq ft of grate per hour

MR. KELLOGG: The chart that Mr. Houghten has just explained to you the Committee feels will prove simple to apply, although somewhat arbitrary. It

will probably be modified in certain particulars as experience in its use dictates, but for the purpose for which its use is suggested, it will prove entirely fair to every manufacturer and heating contractor.

The proposed rating code that you have before you is essentially the same one we offered six months ago, with the addition of the chart.

With this presentation, therefore, the Committee submits the code to the Council and to the Society for its consideration, and recommends its adoption.

PRESIDENT WILLARD: I want to say in connection with the presentation of the Report of the Committee on Code for Rating Low-Pressure Heating Boilers, that the rating code has been under consideration by the Society for almost as long a time as the testing code which we have had as an official code of the Society for a number of years.

The Council has reviewed the rating code, which is an entirely new code, in the last three months. The suggestions made to the Rating Code Committee at West Baden and subsequently, to the West Baden meeting, have been given very careful consideration, and those suggestions have been incorporated, as far as possible, into the rating code that Mr. Kellogg is presenting this afternoon.

The Council is in entire agreement with the report of the Committee. The code itself, like the testing code, undoubtedly will require interpretation from time to time. It will be necessary to have such an interpreting committee, and the identical committee, of course, which will function for the testing code, will, if the Society does adopt the rating code, be charged with the responsibility of interpreting and keeping the rating code up-to-date, bringing into agreement the conflicting ideas and suggestions that may be offered by the members of the Society as they try to apply the rating code in actual practice.

The Council is equally unanimous in its endorsement and acceptance of the rating code, and they recommend the code, unqualifiedly to the Society for adoption at this time.

L. A. HARDING: I move that the code for the rating of heating boilers burning solid fuel as of date January, 1929, be adopted by this organization as one of its standard codes. May I say just a word?

To my positive knowledge, this Society has battled with this situation for a period of sixteen years. I was one of the conscientious objectors at the West Baden meeting to the proposed code as reported at that time. Since that time the Committee, I think, has fully and adequately met many of the objections to the code that were presented at that meeting. I did not feel that the code, as written at that time, fully reflected the physical characteristics of the boiler. I think that the Committee has now met that objection and the two limiting curves reflect two things—grate surface and heating surface in the boiler—and that is why I am in favor of adopting the code at this time.

Now, no codes are perfect; no one could possibly write perfect codes. Changes no doubt will have to be made from time to time to bring it up-to-date. That is true of all codes.

W. H. CARRIER: I want to second Mr. Harding's motion. I think I have a word of explanation to make because I, too, was one of the conscientious objectors. I did not pretend to know much about boilers. I have been educated a lot since; so much so in fact that I realize there is a lot more to be learned.

I do believe, however, that the Committee has given this code a very thorough consideration, and has done a great deal of constructive work on it, also that they have considered various organizations and manufacturers in so far as they could get any agreement or expression of opinion.

Another question in my mind was, do we need any code of this kind at all? The fact that other organizations, whose members use these boilers, have felt that there was so much need of a code of rating as to adopt a tentative one has almost convinced me that some code, one on a sound scientific basis if possible, is desirable.

Our Council has given the subject matter of this code thorough consideration in two lengthy meetings, and, while we do not profess that this is by any means an ultimate or perfect code, it does meet many of the previous objections and forms, at least, a basis on which to start. We are providing an able committee to make such improvements and alterations as may be found necessary to work out the code on a practical basis.

I realize that the effect of a code may be to alter manufacturers' designs. If this is toward the interest of the public, as well as the manufacturer, then such a code is beneficial. On the other hand, it is possible for a code, if its provisions are inadvised and not promptly altered, to work in the reverse direction. We do not as a Society wish to do harm to the interest of any manufacturer nor of the public and, should any such tendency be shown, I can assure you that measures will be promptly taken to have the code properly changed. It is also inadvisable to have any code so rigid that it will tie the manufacturer down and not permit rational and progressive developments.

This code is not intended as a fixed and unvarying law, but one which can now be adopted and changed as the art advances, or as our knowledge of the art increases. In fact, it should be made a very important subject of research at our Laboratory for the determination of the fundamental factors involved and their physical and economic relationship.

I cannot see the argument of those who say, after all these years of effort to obtain a code, to wait still longer until we get a perfect code before it is adopted by this Society. Such a procedure is contrary to all human experience. If the manufacturer waited until he had perfected a new product to place it on the market, it never would be placed on the market for the very good reason that engineers and manufacturers are human beings and not omniscient. A product can only be perfected by being placed in use and its shortcomings and possibilities of improvement determined in actual practice. The same is true of this code, and such is the method this Society must adopt if it ever has a code. It is for this reason that I am seconding this motion and recommending it to you for adoption at this time.

PRESIDENT WILLARD: You know, the discussion on this matter is very important. I would like to have Dean Anderson say a word or two in this connection.

DEAN ANDERSON: Mr. President, I think I have never had anything in all my life that gave me as much concern as this rating code. I must confess that I do not know which way to turn. I have been in a curious frame of mind for months about the whole thing. I was certainly opposed to the code that involved the one principle of trying to rate all boilers from flue-gas temperature of 700

deg. That was so foreign to my intelligence that I was absolutely opposed to it, and without any conference with anybody, and just in a spontaneous moment, I opposed the code as presented at the West Baden meeting.

Now, this Committee accepted the will of the Society at that meeting, and got away from that 700 deg point. They also got away from the question of selection of boilers.

Now, I have gotten myself in this frame of mind in reference to the Boiler Rating Code: we have been jockeying with this boiler rating code for a long time, and I feel satisfied if a document of this kind is too many times turned back to the Committee for revision that it will eventually die in its tracks. Do we want to do that, or do we want to try and put ourselves in a constructive frame of mind, boiler manufacturers, members of the Society in general, and users of boilers—do we want to try the experiment of seeing how illogical this proposed code is, or how good it is?

I have been forced, through the opinions of scientific members of the Society and men who have been giving this house boiler method of analysis a great deal of thought, to come to the conclusion that we can afford to adopt this code, and if it is absolutely wrong, then let the boiler manufacturers, and those who are interested in the design of boilers, show to this Society that the thing is a fallacy from one end to the other, but I would like to see the code adopted as a starting point. I do that because it seems to be the wish of men who are doing the constructive work of the Society, that we should have at this time something to begin to work on. If it is wrong, then it becomes the most virile target that the Society has ever had, and the code will be torn to pieces, and shown in a very short time to be absolutely inadequate for the purpose for which it was intended.

I do not know whether the code is right or wrong. I do not see all the relationships. I am going to be perfectly frank. We may be adopting an arbitrary process of doing something that will never work out. I have a sort of a feeling that we will find many fallacies in this present code presentation, but I am in favor of adopting the code now unless it does somebody some great injustice; if it can be shown here that this code is to do the boiler manufacturers an untold injustice, then I am opposed to it, but I, after thinking the matter over many, many times, have come to the conclusion that this code cannot do any harm because we have protected the Society by putting it in the hands of a competent committee of experts, to discuss the code, and to revise where inadequate or faulty. This continuing committee stands as our official board of experts to receive all objections, and certainly if the code is not right the boiler making world will show to this Society that we have committed the fool act of adopting something that does not work out in our engineering practice.

That is my own personal attitude. In a sense, it is a sort of a confession, that I am not fully cognizant of all of the elements that enter into this code; it is more elastic than the idea of a code limiting the performance of boilers to one of the variables, one making a constant as the guiding star of what boilers can do. In that respect it is certainly more elastic than anything we have had presented to us heretofore.

I know that hundreds of men manufacturing boilers are interested in this.

There has been in the minds of many men—not in my mind—a general im-

pression that there is some subtle and secret process in reference to the design of boilers, that the boiler maker is not ready to give to the world, some curious and weird analysis that we must not know about.

Gentlemen, no progress in all this world has ever been made by any manufacturer who takes that attitude. The attitude of the world today is to let everybody have all the information in reference to everything. Why, you can all remember the time when every manufacturer of machinery, harvesting machinery makers especially, adopted curious threads on all fastenings, so one never could find a nut to make a repair except by sending to the original manufacturer.

And so we went through the throes of labor in bringing forth a standard screw thread which everybody now uses. Now, you young men present can hardly conceive of such a mechanical chaos as that, but there was a time when even the screw thread was as variable as the winds in summer and it was thought good business to make it mysterious. Fix a bolt so that no one could ever put on a nut unless he went first to the manufacturer who had made this particular nut for this particular bolt.

We have seen that sort of evolution go on in every engineering development. The greatest progress that the world has ever seen has been in the automobile business. Why? Because every known variable in connection with the building of internal combustion engines has been given to the world, and for an illustration, look in the lobbies of this hotel just now where you will see that many of these gentlemen have brought their machines and put them side by side to be torn to pieces if faulty. In other words, they want the world to know what is good, and it is the common practice of automobile manufacturers today to take machines of various types and put them under severe tests of endurance. They have laboratories and testing grounds where they will take a well-known machine that seems to be giving service, and drive it under every conceivable difficulty to see where it fails. The manufacturers also do that with their own cars.

The manufacturers look for excellencies in other cars. As a result of all this frankness today we have the most perfect romance of transportation that the world has ever seen, and it has been due, gentlemen, to the fact that all automobile engineers have been willing to give to all the world all the facts.

Now, can we do that in reference to the house heating boiler? I think the boiler manufacturer, if he is a wise man, will adopt this code and say that he believes in the good faith of the Society; and that he believes here is a conscientious attempt to find out what boilers will do and what the difficulty is in the design of boilers. If you will join with us in good faith and try to find out if the proposed code is rational I think this Society will have made very decided progress.

I know that this code is going to be attacked this very day. I have talked with one gentleman at least who is a most eminent scholar, a great authority, Dr. Brabbée. He may not agree with the code as it stands. His theories in reference to boilers and his practical contact with the boiler design and boiler performance is world-wide. But I think Dr. Brabbée can afford to say, "I will accept temporarily this wish of this Society but we will try to show you, in a reasonable time, the fallacy of this report, and through the continuing committee

we will be able to give you some additional information so that within the course of a few years we will have a process of rating which will be invaluable to the profession." Would not this be an idealism in our engineering procedure if Dr. Brabbée would join heartily yet antagonistically in the tryout of the code?

I think furthermore that there will spring up more laboratories for the study of boilers. We have a new laboratory in Kentucky, and the sole purpose of that laboratory is to study the performance of house heating boilers. The one conscientious thing I am going to do as the head of that particular laboratory, or at any rate the man shaping its destinies and its work, is to see what is the matter with this code, and if there is anything the matter with it, through a very eminent scientist named O'Bannon, we will fire you some broadsides before this Society within a year that you will listen to.

Now, if we all put ourselves in such an attitude will we not get somewhere with a boiler rating code? As Mr. Carrier indicates, we must have a base line upon which to survey this great field of boiler design and operation. Is this foundation sufficient for you to build on? Personally I am ready to build on it, having some doubts about it, some serious doubts. Take for instance the item of flue-gas temperature determination. It has been demonstrated time and time again that one cannot measure flue-gas temperature by sticking a thermocouple in the smoke pipe. One gets a variation of sometimes as much as two hundred degrees. That is one procedure that must be more adequately specified.

In determining this curve of flue-gas temperature, there must be some research done on how to measure flue-gas temperature. I have asked Mr. Carrier, this eminent authority on heat transmission, to devise a method for getting flue-gas temperatures which would be substantially a thermocouple so that it would face always a temperature that is practically the temperature of the flue gas. I have asked Mr. Harding to outline a method for obtaining flue-gas temperatures involving the principle of a portion of the flue gas passing through a by-pass path at high velocity. Those two investigations we will take up immediately at the University of Kentucky.

We will check those two methods and see if it is possible to finally establish a standard practice for the measurement of flue-gas temperatures. There is not a man in this room who now knows positively how to measure them. I am satisfied that Mr. Carrier himself could not measure flue-gas temperatures tomorrow—he undoubtedly could eventually if he set that versatile brain of his at work on the problem. Mr. Carrier, I am satisfied that you could not get flue-gas temperatures absolutely by the method given in the code.

MR. CARRIER: Yes, I can get flue-gas temperature.

DEAN ANDERSON: You cannot get it by any method that has been described in any code that we have.

MR. CARRIER: It will give us comparative temperatures which will be pretty close. I mean test our boiler and test another the same way, you will get the same results, they will be off the same amount.

DEAN ANDERSON: Oh, they will not be comparative. Anything that involves a variable error is not comparative. You know that, as a scientist, don't you? Will you admit that variable errors are not comparative?

MR. CARRIER: No.

Dean Anderson: Why, you know well enough that if you have errors that are variable, they are not comparative.

Mr. Carrier: If they are systematic, they are comparative.

Dean Anderson: I do not consider any temperature data systematically taken when there is a variation of 200 deg. If, however, you will do what you said you would about devising a method of measuring flue-gas temperatures and I know you can, because you are the type of thermal scientist who can get the heat emission, oh, let's see—you can get the amount of radiant heat from a mosquito (laughter), but you haven't done it yet. We cannot say to this Society today we must or can get flue-gas temperatures in a certain way. That is for the continuing committee to prescribe later. I use that as an illustration that we do not know how at the present time to get flue-gas temperatures, but we will know how, and that is just one of the points like hundreds of others involved in the evolution of the code.

Are we willing to start out with something that represents a tentative code, and I take it that this is adopted merely as a tentative code and not as a final code by any means—are we ready now after all these years of pulling and hauling, to say, "Well, let's raise this offspring of the Boiler Code Committee?" I think it will all be done in good faith to manufacturers and to the users.

Personally, with all these questions of doubt in my mind, I would be glad to see the Society adopt this simply because it is something from which we can start, and I have no doubt before this discussion ends today, that there will be so many different points raised that we will all be hypnotized, or anesthetized, or something before we leave this room. We will go away from this room with the feeling, well, I think no one knows anything much about a boiler now but some day we perhaps will.

So I confess my ignorance about it, and I am ready to vote for this simply because of the work that has been done by this Committee, primarily because it gives us a chance to start to learn more about boilers.

Just one more thought: I believe this Society adopting a code of this kind is going to start some real discussion on the analysis of boilers, and get away from these mysterious ideas that are abroad in boiler land now. It will do the boiler manufacturers more good than it will anybody else because they will get busy in giving to the American public all the information they can find because certainly they are progressive and in accord with the scientific spirit of the times.

C. W. Brabbée: I would not have taken the floor in this matter because I have found in discussions of the last days that there are great difficulties involved and it might be best to stay away from it. But as Dean Anderson and our President have called on me, what can I do but to respond? I would like to beg you to consider me in this discussion not as connected with any manufacturing concern, but solely as a member of this Society, who has spent a lifetime in investigating radiators, boilers, ventilating systems and accessories, and who wants to serve this Society to the best of his ability.

When I presented but last year, the "Useful Heat Problem" at New York, nobody believed in this idea of rating of radiators, but today, only one year later, we have seen the change.

There is something similar on boilers. We have heard recognized members of this Society talking in favor of this code, and it would be a pity if we would have only one opinion. We must have different ones because only from different opinions real progress can start.

But even those gentlemen who have been in favor of this code have great doubts. Mr. Carrier said he does not know anything about boilers he only learned a little about them in the last weeks. Dean Anderson said he is not sure about these things, and fears something is behind them.

Shall this scientific Society where the leaders are not sure about a thing release it as a law where everybody has to follow the regulations?

Suppose we find this case: a laboratory may be equipped to test boilers according to test code, to which I gladly agree. It starts to comparatively investigate boilers, finds these curves and now proceeds to rate these boilers according to the code. It may be possible—I think it is—that two designers of boilers will make two boilers of the same grate load; yet one boiler may be inferior to the other, inferior in practical use and at the same time may have a lower stack temperature.

I recall a test which was on a different line, but gives you an idea of what can happen. In testing a boiler which ran with 400 deg stack temperature and terrific fuel expense, investigations have shown that introducing secondary air in this boiler, the stack temperature rose from 400 to 800 F but the fuel consumption dropped 20 per cent.

This is surely proof that the stack temperature is not an indication of a boiler's value; just as the condensation was not the determinate factor of a radiator's rating.

If it is possible, that from two boilers running at the same rate, one is inferior in practical operation, and yet has the lower flue-gas temperature, then this rating code will rate this inferior boiler higher than the good one; injustice would be done to the manufacturer of the perfect boiler, an injustice which can never be the intention of us as members of this Society.

Mr. Carrier and Dean Anderson have both said, if such a code will do an injustice to anybody, we will eliminate this code; we will drop it; change it. Gentlemen, if a matter is in such a state of development, then it is not ready for a law, which should be observed.

And there is another point: large manufacturers deal with 90 per cent of customers who don't care for the Society. Is it really possible for a manufacturer to have this large majority of his customers ruled by an arbitrary law of a small minority?

I further question if we, as a scientific body, should enter into the rating of boilers which is and will be a commercial proposition.

DEAN ANDERSON: May I ask you a question? Why is it the boiler manufacturers change their ratings so frequently?

DR. BRABBÉE: Because it is a commercial proposition. Manufacturer A has a rating: Manufacturer *B* comes along and says, "My boiler is rated higher." What can the Manufacturer *A* do? I have been approached on so many occa-

sions and people have told me, if only the manufacturers could give more cooperation, we could do more.

Now, I have, since I am in this great country of yours, always tried to do my part, as small as it may be, to support such cooperation between the manufacturers and the Society, and the universities, and I must say that we have today a much better understanding among those bodies than five years ago.

But, now, in a question which is entirely in the field of the manufacturers, we want to go out and set forth a rule of which we are not even sure if it is right, and enforce it. Wouldn't it be a wise idea to say this: "Here we have a Society, and we have a committee which has worked admirably, honestly, earnestly and made a code, never before discussed with you manufacturers. We want to make this code a rule, but before, we want to have you manufacturers check us up, find out if this will do any injustice to you. If yes, let's correct; if not, work with us."

If a rating code is established without the support of the manufacturers, there is not much use of it. If, on the other hand, it has the support of the manufacturers, its success will be 100 per cent.

May I just tell you a little story? The European automobile industry was crippled by a rating code which could not foresee the development. We just experience in this country laws which cannot be enforced and which are set aside by the power of life, doing great damage. We should be more than careful to keep away from such an embarrassment.

I move that this code may be discussed as an opinion of this Society with the boiler and radiator manufacturers' association, and if they give assurance that this code does not do injustice to them and that they will support it, then make it a rule which will be respected, and which everybody will support. (Seconded by Homer Linn.)

PRESIDENT WILLARD: The Chair recognizes that there are two motions before the house, the original offered by Mr. Harding and a substitute by Dr. Brabbée. Those wishing to discuss the matter will please announce about which motion they are talking.

DEAN ANDERSON: I rise to a point of order. This substitute motion has been made. The discussion must be confined to the substitute motion.

L. A. HARDING: If I may just say this—I am talking on the substitute motion —that the best argument that I have heard for the first motion was given by the speaker that just preceded me.

F. D. MENSING: Gentlemen, the motion before the house as I understand it is the substitute motion that we stop where we are and confer with the manufacturers.

Why limit it to the manufacturers? If we are going to confer, where are we going to stop conferring? For 25 years, I understand from members older than myself, we have been trying to confer with every one. And we are today where we were 25 years ago. I am 50 and I will be 75, and we will be still conferring, and I really would like to see this matter settled before I die; and I think most of us would.

Therefore, I am going to advocate that we stop conferring, do a little

thinking and some quick voting, and that we may clear the board and be able to proceed.

I am going to ask that we have as little discussion as possible on the second motion, and get back to the original question.

S. H. HARPER: This is the first occasion on which I have taken the liberty of addressing the Society but the problem before us has been a serious one with me for something more than 20 years. However, I will try, in accordance with Mr. Mensing's suggestion, to be as brief as possible and shall confine myself specifically to the motion of Dr. Brabbée.

We need scarcely stop for thought to fully realize that nearly all causes of any moment have been sponsored by and put into action by minorities. Majorities, as a rule, are satisfied to leave in the hands of a painstaking few the development and promotion of difficult problems. All of the fanatics of history, whose ideas are now accepted by the universe, at one time had only the support of extremely scant minorities.

The committee, which devised this code, was chosen because they were purely scientists; no manufacturers, no heating contractors were on this last committee —nobody who had anything to sell. They were selected because we feel that there is a type—though maybe not a superior type—of sincerity about a scientist that you do not find anywhere else. His pocketbook is not affected by his decisions. There is nothing insincere about guarding one's pocketbook but these findings are the result of the painstaking effort of scientists. The representatives of the code committee when discussing it in the Pittsburgh Chapter Meeting, in my opinion, virtually admitted that many of their calculations which did not seem rigid enough were for the purpose of giving considerable leeway to boiler manufacturers. Boiler manufacturers feel, with some justification, that there should be no restrictions placed on them which would prevent them from changing design to produce greater efficiency.

I quite agree with Dr. Brabbée, as must every one familiar with boiler construction and operation, that the operation of one boiler may produce a higher stack temperature and still have a higher efficiency than another boiler having equivalent grate area, fuel capacity and heating surface operating on a lower stack temperature. Nevertheless that is in a sense straining the point.

We do know, as a result of experience, that you may expect generally the same efficiency from boilers that are of a particular type. When a boiler is produced which has "genuine merit," whether by a large manufacturer or a small manufacturer, his competitors produce one somewhat similar. Now those boilers, which are almost identical as to grate area, quantity and position of heating surface and method of combustion and heat absorption, should conform to a standard of rating. If, beyond these and other basic requirements covered by the code, a boiler has superiorities of material, workmanship or efficiencies of any nature, such superiorities should be the selling arguments which would enable the manufacturer with the better boiler to sell more boilers or sell his product at a higher price than boilers which did not possess these qualities. At present, and for many years past, inferior boilers are frequently purchased because the purchaser is of the impression that it will perform equally as well as a superior boiler, since both boilers have the same rating. The adoption of a rating code will improve the method of selecting boilers. In sections where

"Certified Heating" has been sponsored by the *Heating & Piping Contractors' Association,* the selection of boilers has been affected very materially by the requirements of these standards. During the past twenty years, with extremely rare exceptions, every manufacturer of any boiler, fired by whatever fuel, has made plenty of money in spite of superiorities or inferiorities of his competitor's product.

It would be an immense satisfaction to us all if this code could have the unanimous support of boiler manufacturers, heating contractors and all other elements represented in this Society. But regardless of the attitude of those who may not be in agreement with the findings of the rating code committee, we have a right to fortify ourselves against the legion of heating contractors, engineers, architects and ultimate consumers who are demanding more insistently every day: "Why the devil—and worse than that—can't you fellows that are selling boilers get together on a standard of rating?" I think I need not add that I shall cast my vote against Dr. Brabbée's motion, and for the previous motion.

THORNTON LEWIS: Speaking on Dr. Brabbée's motion, and referring to one thing that Dr. Brabbée said, particularly when he called attention from his expert experience to two possible boilers that might be unjustly rated according to the code which has been presented, might I just remind all of us that no confusion that can exist under the proposed code could possibly equal the confusion that exists today without any code. Under this code and an interpreting committee there is some appeal, and some remedy. Today, without a code, there is no remedy and a discussion of 25 years has not and I doubt ever will bring a remedy.

The Society may not represent but a very small per cent of the purchasers of boilers. Granted, Dr. Brabbée. That is equally true of every code the Society has ever yet adopted, so I submit that that is not a logical argument, why should we suspend action here and start 25 years of discussion.

Under the rating code proposed, we are not asking manufacturers not to give any rating they want to their boilers. All we are asking them to do is, among all other ratings, to give one known as the A. S. H. & V. E. rating, by which engineers, if they want, may use a common yardstick for comparison. Any other ratings they want to give, they may. They may give a thousand, but at least according to what we know now and the best that we know now, we will have a common yardstick of measurements.

I, therefore, suggest that we defeat unanimously Dr. Brabbée's motion.

PRESIDENT WILLARD: Is there any further discussion on this subject? The Secretary w'll please state Dr. Brabbée's motion.

The motion made by Dr. Brabbée was that the Code for Rating Low-Pressure Heating Boilers as presented by the committee be made the subject of conference between manufacturers of boilers and the Society's committee.

A vote was taken and the motion was defeated.

PRESIDENT WILLARD: The original motion made by Mr. Harding is open for discussion.

R. V. FROST: Since I have been charged with a share of the responsibility in keeping this code alive, I would like to take the opportunity to state my own

ideas. I will give you a little of the history of the development of rating codes.

A few years ago the *Heating and Piping Contractors' Association* came out with a code that was in general form like the chart that you see reproduced on the board.

That code had a great deal of merit. The chart used by the Heating and Piping Contractors was not new for it had been used by some manufacturers before them, but they were the first ones to give it publicity and the idea took among the boiler manufacturers. Perhaps that fact is not generally recognized, but it did take, and I think if the Heating and Piping Contractors had persisted in the development of that particular scheme, they would have settled the rating code sometime ago; and the rating problem.

But due to some kind of a mixup within their own organization, they turned to the key system of rating, which was generally opposed by the manufacturers, and for that reason the manufacturers withdrew their support, and the output committee of the Heating and Piping Contractors, instead of trying to break down the opposition within their own organization, which the members of the committee recognized as wrong, they turned to another system of rating which bases the rating for the square feet of grate area, an idea which I do not believe will persist.

As a result of some of their original charts, some of the manufacturers began to develop performance tables, giving the performance of their boilers over the full range of operation.

That method has been employed by the American Radiator Co., and those of you who visited the Brabbée Laboratory at the meeting last year probably saw several performance tables on the wall, great, large tables you could easily read across the room. I don't know whether those are available for general distribution, but, on request, I believe they can be obtained.

The Richmond Radiator Co. three years ago developed performance tables, giving the full report on the operation of all their boilers. They have had these tables in constant use, and have given wide publicity to the idea. The first issue of their books, ten thousand, was taken up within the first three months; there was a reprinting, and now there is a third reprinting of those same tables. The result of publishing performance tables on their boilers resulted in eliminating almost entirely trouble from improper selection of boilers.

The Code Committee has recognized the value of performance tables, where they say here: "A series of tests shall have been made and a Performance Chart plotted showing—for the purpose of this Code—flue-gas temperature, rate of burning fuel of a calorific value of 12,500 and draft at smokehood against rate of output." Then they make this statement.

"This Code is not intended to supplant the more correct engineering practice of determining the available output of boilers for specified operating conditions by a study or analysis of the complete data given by 'performance charts'; nor does it preclude the assigning of other rating output values to meet purchasing specifications or a specified set of operating conditions. The rating outputs as determined by this Code are intended for average conditions and for the use of purchasers not competent or desirous of making comparisons and selection from performance charts."

In other words, the whole purpose of this rating code is to cater to the incompetent man. Here is a Society based, in its very principles of organization, upon the uplift of the industry, but by this code it is catering to the incompetent man that it wants to drive out of business.

Now, as a matter of fact, there are not so many incompetent men in the heating and contracting business after all, if the experience of the Richmond Radiator Co. is a criterion, for they have found that invariably the contractor in a very short time learned to use their performance tables with the result that there has been an almost total elimination of the trouble from improper selection of boilers.

Now, the boiler and radiator manufacturers, just about two weeks ago, adopted a testing code which bases its result upon a chart that takes the form of the original heating and piping contractors' code. This is a copy of boiler and radiator manufacturers' code, and it can be obtained, I think, from Mr. Herendeen by anyone that desires it.

This code was adopted unanimously by the boiler manufacturers who are members of the association, and the only purpose that the code can have will be to standardize the preparation of performance data and performance tables that tell the complete story of the operation of a boiler.

I believe that is what the industry wants. I know that the manufacturers are opposed to the one-number system of rating. The performance chart and the performance table is really a multi-rating system. You have such a system of rating for fans, for unit heaters, for blast heaters and for motors; the furnace manufacturers have the same type of code; why cannot the boiler manufacturers have one, too? That will be the demand, and that is why we have the opposition to the code in its present form. I imagine that we will have plenty of objections to the proposed code before our new Committee gets very far along in its work.

H. M. HART: I would like to make a slight correction in the impression that might have been given by Mr. Frost, if I may. The *Heating and Piping Contractors' Association* have never abandoned their performance chart. The only reason that we are not using it today is that we never got any performance charts from the boiler manufacturers, with the exception of two, I think, that could be used. We got a few scattering charts on one or two sizes, but nothing complete enough to satisfy the demand of the members of our association, who are trying to operate under the certified heat plan; so we had to adopt something that we could use in lieu of the other, until we had something better. We did not contemplate using the key system as a national association where the performance charts were supplied, but some locals used it. The present net load ratings that we use are, as I say, a necessity because we have nothing better and we will continue to use them until we do have something better.

PRESIDENT WILLARD: You have all heard the motion which was presented by Mr. Harding and the discussion which has just been concluded. All those in favor of the motion please stand.

The Chair rules that the vote is overwhelmingly in favor of the motion, unless those opposed demand a ballot we will not take the time to count the votes. If there is no objection, the vote is declared practically unanimous.

The recommendations made by The Guide Publication Committee in its report were given by Secretary A. V. Hutchinson:

Report of the Guide Publication Committee

YOUR Committee, entrusted with the compilation and publication of THE GUIDE, 1929, have the following to report as to the results accomplished and as to the immediate requirements for the carrying forward of this work in the future.

The Guide, 1929

THE GUIDE, 1929, has undergone the most complete revision of any issue since its inception. All of the chapters except one have been completely rewritten and revised so as to contain the very latest and fullest information available.

In addition nine new chapters have been added covering for the first time the following subjects: Chimneys, Fan Furnace Heating, Central Heating Systems, Heat Exchangers for Water and Oil, Kitchen, Laundry and Hospital Equipment, Pipe and Fittings, Design and Operating Data for the Mechanical Equipment of U. S. Treasury Department Buildings, Selection of Fans for Ventilating, Drying, etc., and a list and Synopsis of the Society's Codes and Standards.

A few of the most noteworthy improvements in the former chapters, in addition to the data contained in the new chapters, are as follows:

Chapter I. The tables on heat losses from buildings were extended and brought more up-to-date, the Infiltration section was completely rewritten and made to include the latest data on steel sash, casement cash, pivoted sash, etc., and the section on Outside and Inside Temperatures was greatly amplified and made to include data on summer as well as winter requirements.

Chapter II. Revised to correspond with present conditions where the older types of cast-iron column radiation are being replaced by the tubular, radiant, non-corrosive metal and fin type radiators and heaters in the form of radiators, recessed radiators, concealed heaters, concealed radiators and cabinet heaters.

Chapter III. Data on newer heat emitting units and many diagrams were added covering typical piping connections.

Chapter V. Better data included on the selection and rating of boilers. Also new data on the erection and care of boilers and on the burning of different kinds of fuels.

Chapter VII. Data on different kinds of grates added and practical data on rates of combustion, air quantities and pressures, and sizes and types of mechanical draft apparatus required.

Chapter XV. Revised to include material on insulating cold surfaces and for the prevention of sweating, also on the determination of economical thicknesses and characters of insulation best suited to various requirements.

Chapter XVI. Much new data inserted on automatic controls for house and hot-water heating and on sectional or zone control for larger buildings.

Chapter XVIII. Changed to conform with latest practices and to include data on traps as well as on pumps for heating and ventilating service.

Chapter XXIII. Data included on the latest practical methods of measurements and the latest laboratory data on the Physiological Effects of Ventilation, including that on the intercharge of heat and moisture between human beings and their atmospheric environments.

Chapter XXXIII. Entirely rewritten to include all of the latest data from the vast amount of research recently conducted upon this subject.

In general a great deal of cross reference work has been done and many explanatory notes and footnotes have been added.

The indexing has been greatly improved and some very useful suggestions for the proper and expeditious use of the material have been added.

The 1929 edition contains thirty-four (34) chapters and 488 pages of text whereas the 1928 edition contained twenty-eight (28) chapters and 366 pages of text.

The catalog data section of THE GUIDE, 1929, contains 324 pages of paid-for manufacturers' data whereas the 1928 edition contained 284 pages.

Statistics

	1928	1929
Income:		
Advertising	$27,460.22	$31,539.94
Copy sales	3,997.49	6,010.28
	$31,457.71	$37,550.22
Expense:		
Printing	$ 9,482.35	$10,788.80
Sales promotion, booklets, etc.	2,505.77	1,077.68
Paper	1,763.71	2,099.39
Mailing	1,408.80	1,586.96
Engraving and art work, electros	645.70	989.47
Miscellaneous	189.19	148.59
Binding and mailing prior year's Guide	408.30	1,348.49
Special compensation:		
Editorial	3,000.00	5,000.00
Solicitation	1,728.00	1,702.08
Traveling	662.71	653.84
	$21,794.53	$25,395.30

Distribution:

8500 copies of The Guide, 1929, were published of which 7000 were bound and the remainder are held in reserve. Distribution up to December 31 was as follows:

Members American Institute of Architects	2615	2643
Members A.S.H.&V.E.	1954	2010
Members Heating and Piping Contractors' Natl. Association	682	859
Advertisers and their agents	260	280
Copies for sale	2789	2708
	8300	8500
Size:		
Pages of advertising	312	356
Pages of text	368	488
	680	844
Copies Printed:		
Total copies printed	8300	8500
Total copies bound	7000	7000
Cost per copy	$2.626	$2.987
Income from Advertisement per copy	$3.475	$3.738

This Year's Organization

This year's edition was handled under a general publication committee consisting of the General Chairman and Editor-in-Chief and two Vice-Chairmen.

Under this central organization there were thirty-four (34) committees, each headed by a specially selected chairman, with from five (5) to ten (10) members for handling each section of The Guide.

In addition there was an Advisory Board consisting of about twenty-five (25) members for passing upon special subjects and questions.

The work has thus been spread among over two hundred (200) members located throughout different parts of the country and affiliated with the various Chapters of the Society.

In addition to our own members, several of the best authorities on some of the subjects, who are not members of our Society, have contributed. Among these are D. J. Bergman, who handled the Chapter on Heat Exchangers for Water and Oil, and George A. Orrok, who handled the Chapter on Chimneys.

Under the organization this year cooperative work was started with the *Heating and Piping Contractors' National Association* and others for the coordination of data on all subjects connected with the estimating of heating requirements.

In cooperation with Chairman Lewis and the Research Laboratory at Pittsburgh several investigations have been inaugurated for the collection of data needed for the future improvements of The Guide. These included: tests on return pipe sizes, especially for systems with thin copper blast heaters, test data on infiltration of the casement and pivoted type of steel sash, both with and without roll screens.

The Secretary's office and staff have contributed an unusual amount of work, including the entire promotion and sales of the catalog data space, the entire manage-

ment and handling of the printing, redrawing and arrangement of cuts, indexing, proof readings, corrections, binding and mailing.

Mr. Hutchinson personally has contributed much time and pains in arranging for and attending committee meetings, on coordination with the other Society publications, on final arrangement and printer's instructions, and on general make up. J. E. Bolling has acted as consultant and chief adviser in the matter of preparation of catalog data and distribution of THE GUIDE.

We take this opportunity of again calling attention to the vast amount of unselfish and invaluable work contributed to this publication by the various Chairmen and their Committees, who have performed the real work of collecting and compiling the data for the text section. As is stated in the preface of this edition, there is no way in which the greater number of these men could have been induced to expend the time and thought which have been so lavishly contributed to this edition, except from the motive of unselfish service to such an unselfish and useful cause.

The names of the men who have so devotedly served you, and through you the world at large, are given in footnotes at the bottom of the first page of each chapter. We suggest that you refer to these when using THE GUIDE to recall to your memory the names of these splendid contributors as well as the high authority of the data being used.

The Guide Publication Committee again thanks these men and their assistants for their contributions.

Suggestions for the 1930 Guide

We would recommend that the present form of organization be continued with the exception that the General Chairman have the assistance of a full-time paid member on the staff of the Secretary and Manager of Publications. Also that the functions of the Advisory Board should be used to a greater extent.

In connection with the various chapters the following items require attention:

Chapter I. The transmission coefficients need extension to more types of construction, especially roofs, floors, cinder fills, and frame construction with and without building paper. The whole list of transmission coefficients and tables should, if possible, be arranged so that they may be adopted by other organizations as a uniform standard in calculating heat losses.

The transmission coefficients for some of the insulating materials need checking and these factors should, where possible, be grouped under certain classes with a certain amount of tolerance each way instead of trying to give the individual factor for each to the second decimal place, as this is only causing controversy and confusion over contentions that THE GUIDE shows one material so much superior to another, because there may be a few hundredths of a Btu difference in the tables, although the actual difference in a composite wall might amount to little or nothing.

The factor for air spaces needs checking and revising for different conditions.

The questions of inside and outside temperatures, guarantee temperatures, base temperatures and exposure factors need to be followed up in connection with other organizations.

The factors for glass under different wind velocities need checking.

Chapter II on Radiators and Heaters, needs to have much more data added on the rating and heat emission of all of the newer type of radiators, heaters, cabinet heaters, etc., as soon as this can be made available. Cooperative work on this should be followed up with the work of our Laboratory along these lines.

Chapter III on Steam Heating Systems and Piping needs further enlargement along the lines of the latest specially controlled vacuum systems and along the lines of zone and sectional control of systems for larger buildings.

Chapter IV on Hot Water Heating Systems and Piping needs more data on forced circulation systems, comparative costs, economies and simplification throughout.

Chapter XVI on Automatic Heat Control needs enlargement along the lines of more data on zone and sectional control for larger buildings.

Chapter XXIII on Ventilation needs to be further coordinated with latest investigations and practical applications.

Chapter XXX on Ozone in Ventilation needs to be extended along the lines of the latest practical advantages and application of ozone to specific problems.

Generally

The cooperative work started with the *Heating and Piping Contractors' National Association, Plumbing and Heating Industries Bureau, American Gas Association, National Association of Master Plumbers* and the *National District Heating Association* should be carried through to successful results.

The research work inaugurated at our Laboratory on Return Pipe Sizes, Infiltration, and for determining the proper air conditioning and ventilation standards for summer time ventilation should be followed through to conclusions for practical results in future issues of THE GUIDE.

The work of the Standing Committees and the Committees which have been continued should be followed up as expeditiously as possible, so as to have the data ready on or before June 1, 1929. We should also make wider use of the other Committees of the Society.

The undertaking to induce manufacturers to include more of their data in THE GUIDE so as to eliminate a lot of heterogeneous catalogs and to standardize on our page size so as to make their catalog data and cuts fit into THE GUIDE should be followed up.

Lastly, the work of compiling THE GUIDE, 1930 should be organized and well under way immediately following the Annual meeting with the end in view of having all data ready for editing not later than June 1, 1929, and edited ready for the printers not later than August 1, 1929. THE GUIDE should come out in October instead of December.

Respectfully submitted,

PERRY WEST, *General Chairman*
W. H. CARRIER, *Vice-Chairman*
C. V. HAYNES, *Vice-Chairman.*

President Willard pointed out that in 35 years of the Society's life, it had had the opportunity to offer honorary membership to four men, namely, Wm. J. Baldwin, Dr. J. S. Billings, C. W. Newton and John Gormly. Several years have passed since the Society has formally recognized men in this caliber and at its last meeting the Council received the proposals and recommended and received favorable action in the case of two men, in accordance with the provisions governing honorary membership, covered in the Constitution and By-Laws, ARTICLE III, *Section 1.*

The first nominee was presented by S. R. Lewis, Chairman of the Committee on Research, who said, "it seems fitting that the name of O. P. Hood, chief mechanical engineer of the U. S. Bureau of Mines, should be presented by unanimous approval of the members of the Committee on Research, for Mr. Hood was instrumental in establishing our Research Laboratory and it was through his interest and enthusiasm that a cooperative agreement with the government was established."

The motion of Mr. Lewis for a favorable vote on the nomination of O. P. Hood for honorary membership was seconded by E. C. Evans of Pittsburgh and unanimously carried.

Thornton Lewis, Philadelphia, in presenting the second candidate for honorary membership, stated that a few men with vision caused the Society to come into existence and these charter members had a temporary organization, and elected officers. "Stewart A. Jellett of Philadelphia was the first regularly elected President," said Mr. Lewis, "and when our thoughts turn to the men who have done a great deal for this profession, they should turn to the man who

first headed this organization and it is a great privilege to present his name to you today, having been proposed by some of the men who originally met 35 years ago. Those who signed this petition are A. C. Edgar, Philadelphia, a charter member, who has attended every Annual Meeting of the Society, J. J. Wilson, a charter member over 80 years of age, who never fails to attend the Philadelphia Chapter meetings, George W. Barr, F. D. Mensing, R. V. Frost, C. V. Haynes, George D. Hoffman, J. D. Cassell, H. P. Gant and Thornton Lewis."

The motion of Mr. Lewis to elect S. A. Jellett an Honorary Member of the Society was seconded by E. S. Hallett and unanimously passed.

The Honorary Membership Certificates for O. P. Hood and Stewart A. Jellett carry the following citations:

> The AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS hereby confers Honorary Membership upon O. P. Hood in recognition of the debt which the arts and science of heating and ventilating engineering owes him for his never failing interest, encouragement and support in establishing and maintaining the Society's Research Laboratory.
>
> As a distinguished engineer in public service and as an author and educator the Society desires to commemorate his eminent service.
>
> The AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS hereby confers Honorary Membership upon Stewart A. Jellett in recognition of the debt which the arts and science of heating and ventilating engineering owes to him.
>
> As a Charter Member and the Society's First President, as an engineer of outstanding attainments and because of his many contributions to the up-building of the heating and ventilating engineering profession, it is desired to commemorate his accomplishments and his service to the Society.

President Willard asked the Secretary to read the amendments to the Constitution and By-Laws and the members present discussed the proposed changes in the Constitution and By-Laws which were prepared by a special committee composed of W. H. Driscoll, H. P. Gant and F. R. Still, in accordance with a resolution of the Council dated March 28, 1928.

On motion of Mr. Langenberg, seconded by Mr. French, the amendments to the constitution were unanimously adopted as follows:

Amendments to the Constitution and By-Laws

CONSTITUTION

(1) ARTICLE II, *Section 7,* reads as follows:

> "A Student Member shall be a person under twenty (20) years of age who has had insufficient experience to qualify for a higher grade of membership."

to be amended as follows:

> "A Student Member shall be a person over sixteen (16) years of age who is regularly attending day courses in a college or engineering school."

(2) ARTICLE III, *Section 3,* reads as follows:

> "When a Student or Junior Member reaches the age limit of his grade, his membership has automatically terminated unless he be transferred to a higher grade of membership, as provided in the By-Laws."

to be amended as follows:

> "When a Junior Member reaches the age limit of his grade, he shall be automatically transferred to Associate grade of membership, as provided in the By-Laws, unless he has applied for and been elected to Member grade."

(3) ARTICLE III, *Section 4,* to be added:

"When a Student Member discontinues his regular day studies in a school or college, it shall be incumbent upon him to apply within one year for advancement to either Junior, Associate or Member grade."

The amendments to the By-Laws were then considered as a group and practically the entire discussion was confined to Item 7, the amendment to ARTICLE XIII of the By-Laws.

BY-LAWS

(4) ARTICLE III, *Section 6,* reads as follows:

"Junior Membership shall be limited to eight (8) years and upon election to a higher grade, such members shall upon notification of transfer, pay an additional initiation fee of five dollars ($5.00) and thereafter pay the annual dues of the grade to which they are transferred."

to be amended as follows:

"Junior Membership shall be limited to eight (8) years and upon election to a higher grade, such members shall upon notification of transfer, pay an additional initiation fee of three dollars ($3.00) and thereafter pay the annual dues of the grade to which they are transferred."

(5) ARTICLE III, (*New*) *Section 7,* to be as follows:

"Student Membership shall be limited to the duration of regular attendance at day courses in a school or college, and upon election to a higher grade, such members shall upon notification of transfer pay an additional initiation fee equal to the difference between five dollars ($5.00) and the initiation fee of the grade to which they have been transferred, as provided in Section 1 of Article III. Thereafter they shall pay the annual dues of the grade to which they have been transferred."

(6) ARTICLE III, *Section 7,* to be changed to Section 8.

(7) In accordance with Article XIII of the By-Laws, relating to Amendments, the following amendment has been submitted:

"That the Council be empowered to amend *Article III, Section 3,* of the By-Laws in any way that it may see fit in order that *Article III, Section 3,* may comply with United States Postal regulations relating to membership subscriptions, paid for publications entering the mail as second class matter and that when the Council has amended this By-Law, it is to become operative on a date to be fixed by the Council."

DISCUSSION

THORNTON LEWIS: There is one amendment particularly, that I want to talk about at this time, namely, the last one. That amendment is a good one and is necessary in order to give the Council the opportunity to function as the business managers of the Society, and at this particular time there is a very definite reason for passing that amendment.

A very short time ago the officers of the Society were faced with the following situation: Our JOURNAL which has been published continuously since 1915 is now slightly more than self-supporting, due to the advertising in its pages.

One older publication in the field, the *Heating and Ventilating Magazine,* has been directly competing with the Society for advertising. That magazine was sold recently and became one of a large combination of trade papers. Naturally any good business man can see that by such a move, the new owners expect to take a somewhat more aggressive attitude in the field and go after all the possible advertising.

Official announcement had also been made of the entrance of a new publication in the field—*Heating, Piping and Air Conditioning.*

The officers of the Society had in mind, the fact that we have been asking the manufacturers to support several different causes; 1st, catalog data in THE GUIDE; 2nd, we have been asking them to contribute liberally to our Research

Laboratory. In addition, we have been and are now asking these same manufacturers for advertising in the JOURNAL.

Under these conditions, if we could keep from putting our hand out for money to the manufacturers even in one of these causes, there was some advantage. If, at the same time we might cause the influence of this Society to reach to broader fields, that seemed to be an advantage.

If, however, we were going to continue the publication of the JOURNAL in business competition with the older magazine under new ownership, and a new publication to enter the field, frankly, it was believed that the JOURNAL would have less opportunity as a money maker.

So after considering all of the factors involved, an agreement was negotiated between the Society and the new publication, *Heating, Piping and Air Conditioning* on the basis of a specific proposal made by the publisher to incorporate our JOURNAL as a separate section of the new magazine. Informal inquiry was made of an official of the *Heating and Ventilating Magazine* as to his opinion of the desirability of having an established paper enter into an arrangement to act as publisher for an organization. As no offer was made by the Society nor received from the publishers of *Heating and Ventilating Magazine,* it was not considered by the Council.

Briefly, the plan for the JOURNAL provides that instead of being published separately and on rather poor and weak financial legs, it would be incorporated into a larger and more imposing publication as a separate section with considerable financial advantage to the Society.

Now, this was a matter that the Council would have preferred to bring to this annual meeting and discuss before any action was taken by it. The Council, under the constitution, has full authority to handle all business matters of the Society; and in this instance, it had to act; it could not wait. Without going into many details, it was simply impossible, if such an arrangement was entered into, to wait for this meeting.

The Council deliberated quite at length over this move. It recognized that there is a sentimental value in our publishing a JOURNAL separately, but after considering all the factors involved, it decided that the best interests of the Society, from every standpoint, not financial alone, impelled it to accept the proposal made.

The matter was, therefore, placed in the hands of a Committee, consisting of Mr. Jones, the chairman of the Finance Committee, and the speaker. Contracts that were drawn were submitted to an attorney employed by us who had had a good deal of experience in the publishing field, and another attorney for one of the large manufacturers in this industry, with which the writer is not at all connected, who is a man with a great deal of experience in handling New York State contracts.

These two attorneys approved the form of agreement which has finally been signed by your President and Secretary.

Under the terms of this agreement—and all members have received a notice to this effect—THE JOURNAL will become a part of *Heating, Piping and Air Conditioning.* As near as I can give you figures on it, my judgment is that this agreement, if it is allowed to run its full course, will net the Society an

additional ten thousand dollars per year as income over that which it is getting now, and perhaps several thousand dollars more if we consider what it would get if our own JOURNAL continued.

The contract is for a period of three years. At the end of that time, the Society alone has the right to renew it for an additional two years. There is a sliding scale of return to the Society, and it is a little complicated arrangement because we also have to subscribe to THE JOURNAL for our members, but when it is all boiled down, what it means is this: About ten thousand dollars additional income a year to the Society.

Now, we might not want to do this if it were purely from a financial standpoint, but it is supposed this journal will reach a very large number of people in this field that our JOURNAL does not reach, probably double the number, or maybe triple the number.

That, we feel is a big advantage, more people in this industry are going to know what this Society stands for, the kind of work it is doing, and eventually we will receive more support for research than we are now receiving.

We will undoubtedly get many new members as a result of them and so from every standpoint the Council has felt that this was a good move.

I am sorry there are not more members here at this session because I would like every man in the Society to hear these words of explanation so they might know the reasons which prompted the Council to act and act quickly.

If there are any questions that any members would like to ask, I should be glad to answer them. It is absolutely necessary that we pass the last by-law, on your printed sheet, in order that this contract may become effective.

PRESIDENT WILLARD: I wish to thank Mr. Lewis for this lucid explanation of this action on the part of the Society, and I wish to say that the Council is in thorough accord with this matter. There are, of course, arguments on both sides of this question, but the arguments put forth by Mr. Lewis have unanimously convinced the Council that this is a business venture, a logical advancement of the dissemination and the influence of this Society and that it was good business.

E. K. CAMPBELL: I would like to ask if the Society still maintains its editorial staff.

MR. LEWIS: The Society has full editorial control. The publication cannot change one word of the copy that goes into our section, without our consent unless such matter should be libelous.

We retain full rights over everything that goes into our Section. We retain full rights if at any later time we want to publish our own JOURNAL. We have not sold THE JOURNAL; we are simply receiving compensation for the editorial matter that goes in our section of this new magazine. We have tried to be very careful to safeguard the Society, should this arrangement not meet with the approval of the members after being tried over the three-year period.

PRESIDENT WILLARD: Will you state further in this connection that the section in this magazine is a separate unit and under no circumstances can features be taken out of that and played up in the other part?

MR. LEWIS: Not without the consent of our Publication Committee.

Our JOURNAL will be larger, printed on nicer paper, in a separate section, so our members not only get all the things that appear in our JOURNAL but also the benefit of everything else that appears in the magazine, without any additional cost to them. They are getting a little more for their money in that way.

J. H. KITCHEN: Mr. Lewis has made a very clear explanation of the proposition and has evidently given it a great deal of thought and study.

My first reactions were against combining THE JOURNAL with any other paper, no matter how meritorious it was. It has been presented in such an able way, and given a very convincing argument. You haven't completely changed my mind. I feel that it is unwise and the Society will eventually regret going into any other publication.

MR. LEWIS: There is one thing I overlooked. One of the great technical societies in this country is somewhat a precedent for this move, the Chemical Society, where the situation is handled by a private publication company, publishing their journal. Our fellow British Society is handled in a similar way, and I think there are one or two others in this country, so that there is, if we are interested in precedent, lots of precedent for this, among other high grade scientific bodies.

JOHN HOWATT: I was talking last night with a number of members, and the expression was made that the most valuable reference library we have are our bound TRANSACTIONS, and I wish Mr. Lewis would tell the members of the Society here how we will continue to receive our bound TRANSACTIONS in the future.

MR. LEWIS: Exactly the same as you do now. In fact, when I have talked dollars profit to the Society, I have had in mind that that is net after the cost of publication of TRANSACTIONS. In addition to that, the publishers of *Heating, Piping and Air Conditioning,* will furnish the reprints for our meetings. They also agree to keep on hand a certain number, the number to be designated by us, of their monthly publications, so that copies can be bought at a later date by our members just the same as you do now.

Mr. Hutchinson, who has and deserves the greatest credit for working out the details of this proposition, has been very careful to see that there isn't a single right, privilege or opportunity that our members and our Society enjoy today that won't be at least as great and most of them enhanced under the new arrangement, and I think if you will be kind enough to give this idea a fair trial, and keep an open mind on it until you really see how it works out, you will all approve it.

ROSWELL FARNHAM: Just one question: Will you state whether it is going to be the same size TRANSACTIONS issued at the end of the year as before?

MR. LEWIS: Same size. This contract also contains this provision, that *Heating, Piping and Air Conditioning* agree that their publication will not be the official publication of any other organization or Society. So there is no danger of some other organization being in there as part of it, and more or less pushing ours into a small position.

J. M. ROBB: I just want to express perhaps a little different angle of viewpoint and I am very glad that Mr. Kitchen raised the question that he did that seems to relate between the commercial side and what I might call the professional side of our work.

I have been distressed for sometime by the apparent attitude of many people that I deal with, that if a man has something to sell, he necessarily is less honest than if he gives professional service.

Now, it seems to me the time has come when folks have to do a little more straight thinking along that particular line. It all comes back not to the label a man wears, but to his character, and I think while I agree with the view that Mr. Kitchen first expressed, that it was perhaps a mistake to give up our JOURNAL. The more I think it over and particularly in the light of what Mr. Lewis has explained, the more I think that it has been a wise move. Everything that you do has to rest on an economic basis. If the economics are sound, then the movement is wise. If they aren't sound, then experience will tell us so, and that thing applies to your engineering, to your selling and it applies to everything else.

PRESIDENT WILLARD: This last discussion under the general subject of by-law amendments has been confined almost entirely to the last item under the by-law section, and you may wish to vote upon these separately. Is there any wish to separate the amendments?

As a negative response was heard President Willard called for a vote on the group amendments to the By-Laws as printed.

On motion of Mr. Rowe, seconded by Mr. Hart, a unanimous vote in favor of the amendments was recorded.

H. M. HART, Chicago, proposed the following resolution:

> *Resolved,* that an expression of appreciation for the fine work that has been done for the Society by the officers during 1928, by the Headquarters staff, Committee on Research and the Laboratory staff, be transmitted to them so that they may know of the feeling of the members in reference to the service they have so generously rendered during the year.

This motion was seconded and passed unanimously by rising vote.

On motion of Dean Anderson, seconded by J. M. Robb, the following was voted:

> *Resolved* that the Society express by a rising vote of thanks, the appreciation of the membership for the work of the Secretary, whose energy and ability to cooperate with the members of the Illinois Chapter has made possible the greatest meeting of the Society.

Roswell Farnham proposed the following motion:

> That a vote of thanks be given to the members of the Illinois Chapter and the Committees which cooperated so splendidly in handling the details of the 35th Annual Meeting of the Society, and tell them how much the members appreciate the entertainment and hospitality that they provided.

Roswell Farnham, Buffalo, also proposed the following resolution which was seconded by J. M. Robb, and unanimously passed:

> *Resolved* that the President-elect write an expression of the Society's appreciation to the University of Illinois for its cooperation and generosity in permitting the Society to have so much of the time of such valuable men as Prof. A. C. Willard and Prof. A. P. Kratz, and be it further resolved that we express our thanks to President D. Kinley for the splendid talk which he gave at the 35th Annual Banquet.

W. T. Jones, Boston, stated that he would like to present in the form of a motion, an expression of opinion heard from many men attending the meeting:

Resolved that the thanks of this Society be extended to the Committee on Testing and Rating Low Pressure Boilers for the patient, careful and thorough work it has completed, and be it further

Resolved that the President of the Society be requested to express in writing to the individual members of the committee the appreciation of the Society for the many months of fine work they so splendidly performed.

Motion was seconded by H. M. Hart, Chicago, and unanimously carried.

E. K. Campbell, Kansas City, proposed the following motion:

Resolved that the Society in 35th Annual Meeting assembled express to the Edgewater Beach Hotel and its managers, Mr. John Dewey and Mr. B. B. Wilson, some feeling of our appreciation for the service that they have rendered, and the splendid hospitality that they have given.

The motion was seconded by Mr. Robb and unanimously carried.

Roswell Farnham, Buffalo, offered the following resolution, which was seconded by Mr. Robb and unanimously carried:

Resolved that the members of the Society extend a vote of thanks to the speakers who contributed from their knowledge, the splendid papers which have been presented at this meeting, and to advise the Chicago Association of Commerce of our appreciation for the inspiring address delivered by Mr. Oliver J. Prentice;

And that *it be further resolved* that the Society thank the members of the Trade Press for their attendance and the publicity which they have given to the Society's activities.

PROGRAM 35TH ANNUAL MEETING

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
EDGEWATER BEACH HOTEL, CHICAGO, ILL.

Monday, January 28, 1929

10:00 A.M.	Council Meeting
11:00 A.M.	Meeting for Nominating Committee
12:30 P.M.	Luncheon for Authors and Officers
2:00 P.M.	FIRST SESSION—
	Address of Welcome, Oliver J. Prentice, Chicago Association of Commerce
	Greeting by President A. C. Willard
	Report of Tellers
	Amendments to Constitution and By-Laws
	Heating with Steam Below Atmospheric Pressure, by C. A. Thinn
	Report of Council
8:00 P.M.	Meeting of Committee on Research
8:30 P.M.	Informal Reception and Dance

Tuesday, January 29, 1929

9:30 A.M. SECOND SESSION—
Report of Secretary
Report of Finance Committee
Investigation of Heating Rooms with Direct Steam Radiators Equipped with Enclosures and Shields, by A. C. Willard, A. P. Kratz, M. K. Fahnestock and S. Konzo
Experiments on the Effect of Surface Paints on Radiator Performance, by C. H. Fessenden and Axel Marin
Architectural Aspects of Concealed Heaters, by J. H. Milliken and H. C. Murphy
Report of Membership Committee
12:30 P.M. Ladies Luncheon (Marshall Field's)

2:30 P.M. THIRD SESSION—
Report of Committee on Research
Thermal Resistance of Air Spaces, by F. B. Rowley and A. B. Algren
Additional Coefficients of Heat Transfer as Measured under Natural Weather Conditions, by F. C. Houghten, Carl Gutberlet and C. G. F. Zobel
Air Infiltration through Various Types of Brick Wall Construction, by G. L. Larson, D. W. Nelson and C. Braatz
6:30 P.M. Past-Presidents Dinner

Wednesday, January 30, 1929

9:30 A.M. FOURTH SESSION—
Heat and Moisture Losses from the Human Body and Their Relation to Air Conditioning, by F. C. Houghten, W. W. Teague and W. E. Miller
Cooling and Humidifying of Buildings, by S. C. Bloom
The Summer Comfort Zone: Climate and Clothing, by C. P. Yaglou and Philip Drinker
Low Humidity Psychrometric Charts, by M. C. W. Tomlinson
12:30 P.M. Ladies Luncheon and Bridge

2:30 P.M. FIFTH SESSION—
Report of Committee on Rating Low Pressure Heating Boilers, by Alfred Kellogg, Chairman
Frost and Condensation on Windows, by L. W. Leonhard and J. A. Grant
Report of GUIDE Publication Committee
7:00 P.M. Annual Banquet and Dance—Speaker, David Kinley, President of University of Illinois

Thursday, January 31, 1929

9:30 A.M. SIXTH SESSION—
Discussion on Correlating Thermal Research
Report of Committee on Heating and Ventilating Garages
Application of Oil Burners to Various Types of Domestic Heating Systems, by J. H. McIlvaine
Laundry, Kitchen and Hospital Equipment, by H. C. Russell

2:30 P.M. SEVENTH SESSION—
Ratio of Opening of Fan Performance in Terms of Direct Pressure-Quantity Relations, by G. E. McElroy
Installation of Officers
Resolutions
4:30 P.M. Council Meeting

COMMITTEE ON ARRANGEMENTS

H. G. Thomas, *General Chairman*

Finance Committee: August Kehm, H. M. Hart and John Howatt

Transportation Committee: J. H. O'Brien, J. J. Haines and C. W. Johnson

Hotel and Banquet Committee: H. G. Thomas, T. H. Monaghan and E. P. Heckel

Registration Committee: C. W. DeLand, J. F. Hale and G. H. Blanding

Entertainment Committee: J. A. Cutler, A. B. Martin and L. L. Narowetz

Publicity Committee: C. Presdee, R. B. Hayward and E. Mathis

Meetings Committee: H. R. Linn, J. H. Milliken and O. W. Armspach

Reception Committee: W. H. Chenoweth, R. A. Widdicombe, R. L. Gifford, F. W. Powers, J. H. Davis, J. M. Stannard, G. H. Mehring, W. A. Pope, John Boylston, C. D. Allan and C. F. Newport

Ladies Committee: Mrs. J. A. Cutler, Mrs. E. P. Heckel, Mrs. J. H. O'Brien, Mrs. T. H. Monaghan, Mrs. J. F. Hale, Mrs. J. Howatt, Mrs. H. G. Thomas, Mrs. F. H. Gaylord, Mrs. J. J. Haines, Mrs. C. W. DeLand, Mrs. H. M. Hart, Mrs. J. H. Milliken, Mrs. R. B. Hayward, Mrs. August Kehm and Mrs. Eugene Mathis.

No. 820

LAUNDRY, KITCHEN AND HOSPITAL EQUIPMENT

By H. C. Russell,[1] Chattanooga, Tenn.

MEMBER

Introduction

THE requirements of laundry, kitchen and hospital equipment vary widely, *first,* because there are so many different classes of laundries, kitchens and hospitals and *second,* because the operating technique is rarely the same.

Equipment for any of the above mentioned which has been perfectly satisfactory may suddenly become absolutely unsatisfactory when a new laundryman, a new cook or a new set of nurses have been employed.

The data and methods given herein are for average operating conditions, but special consideration should be given unusual requirements.

Piping systems for laundry, kitchen and hospital equipment should have separate flow and return lines, the former of which may be either above or below the equipment; the return lines, however, should be located below.

Traps (preferably thermostatic) are placed on each coil or unit, and the condensation is led into a return main which is itself vented, and proceeds by gravity into a vented receiving tank.

The vertical pipe in connections, especially the drop lines from overhead steam mains, are often dripped through separate traps although they may be omitted as the fixture coils and traps will care for a limited amount of condensation when the fixture valves are opened up without any damage ensuing.

The steam mains should be dripped and drained in about the same manner that would be required in steam piping for any other purpose.

Laundry Equipment

It is not proposed to discuss custom laundries here, but rather, the flat-work laundries which are found in hotels and institutions. Custom laundries are seldom designed to meet certain, definite requirements as in a hospital having so many beds or a hotel of a large number of rooms; they are often built to represent an investment, to meet special requirements in a general way, such as serving one type of population. In hotels and institutions, flat-work laundries are found which have more definite requirements, both as to volume of work and

[1] Inspection Engr., Office of Supervising Arch., U. S. Treas. Dept., Chattanooga, Tenn.

Presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Chicago, Ill., January, 1929.

character. Most of the linen in such institutions consists of flat work such as: sheets, towels, pillow-cases, blankets and table linen.

A great variety of special machines, used mostly in custom or general laundries will not be mentioned here. Some of these are occasionally used in flat-work laundries where a limited amount of other classes of work is to be done but usually, they occupy little floor space and use small amounts of steam, water and electricity. When needed, they may be added without materially affecting the layout of the other and larger machines requiring more floor space, steam, water and electricity. This class includes washing machines, extractors, tumbler dryers and flat-work ironers.

Some notable changes have taken place in the laundry industry during the last decade. The engine and line shaft which were used for many years have been replaced by individual motor-driven machines; the former reversing mechanism of washers has been substituted by reversal of rotation of the motors driving them. In such a laundry, the electrical mechanism for reversing the motors, the switches and rheostats are mounted on a central control panel; these remote control devices are operated by a push switch at the machine.

The reversal mechanism of washers and tumbler dryers is so arranged that two or more machines controlled from the same board cannot be reversed simultaneously in order to flatten out the load curve on the feeders. This is important, especially when alternating current is used, for such motors take nearly twice their running current while starting. Ordinarily, each washer is reversed every eight seconds.

With remote control, a great deal of tampering of the switching mechanism has been eliminated. The amount of exposed gearing and energized electric wiring has also been reduced to a minimum, thus doing away with a great many accidents.

Proportioning Equipment

Probably the minimum equipment of a well-equipped flat-work laundry, the capacity of which is beyond the domestic plant, is comprised of the following: one 36-in. by 48-in. washer, one 36-in. by 64-in. washer, one 26-in. centrifugal extractor, one four-roll 90-in. flat-work ironer, one 30-in. by 42-in. drying tumbler, one two-truck dry room, one 15-gal. starch cooker, one 38-in. pressing machine, one hand-ironing board, two 8-lb electric irons, two laundry tubs and one wringer.

In a hospital laundry one 36-in. by 48-in. by 84-in. disinfector is usually installed, which is large enough to receive an ordinary hospital mattress without distorting it. The disinfector is placed in the laundry in order that the clothes may be sterilized before being washed.

Such a laundry would meet the requirements of a 250-bed hospital or a 350-room hotel on a basis of 44 hours per week, although for a smaller number the equipment would be about the same but operated fewer hours per week. In the minimum laundry, the prime purpose of the two washers is to provide for breakdown service. Washers require repairs oftener than any other machine in the laundry and are indispensable to the operation of the plant. It has been found desirable to have one small washer installed, regardless of how many larger ones are installed.

A few typical cases, which have proved ample but not extravagant, are cited.

The demands of a 750-bed tuberculosis hospital were met by the following equipment, on a basis of 44 hours operation per week:

One 36-in. by 48-in. washer.
One 36-in. by 64-in. washer.
One 30-in. and one 26-in. extractor.
One 42-in. by 72-in. drying tumbler.
One 120-in. five-roll flat-work ironer.
One 36-in. high 48-in. wide and 84-in. long disinfector.
One two-truck dry room.
Four hand ironing boards.
Six 8-lb electric irons.
Four hand laundry trays.
Four wringers, mounted on laundry trays.
Two 15-gal starch cookers.
Two 60-in. long starch tables.
Three 38-in. pressing machines for uniforms, etc.
One 60-gal soap tank.

The flat work for a system of dormitories in which 1800 women were housed and fed was done in a laundry with the following equipment, on a 44-hour per week basis:

Seven 36-in. by 64-in. washers.
Five 26-in. and 30-in. extractors.
One five-roll 110-in. flat-work ironer.
Two 42-in. by 72-in. tumbler dryers.
One six-truck dry room.
Four 15-gal starch cookers.
Four 60-in. long starch tables.
Two hand-ironing boards.
Three 8-lb electric irons.
Two hand laundry trays.
Two wringers, mounted on laundry trays.
Three 38-in. pressing machines for uniforms, etc.
Three 49-in. pressing machines for uniforms, etc.
Four 60-gal soap tanks.

Washing Machines

The washers used in these instances were of the standard type, and about one cycle or washing per hour was accomplished.

A "high duty" washer, as it is called, makes it possible to approximate two cycles per hour, and differs from the standard type in that it has nearly twice as many supplies and waste connections and thus fills and discharges more quickly. In order to realize the advantages claimed for these machines, however, it is necessary that the machines be charged and discharged with corresponding rapidity.

The "sterilizing," another type, has a steam-tight body, operating with a pressure of about 30 lb steam while washing. Although more expensive in initial cost and upkeep than other types, this washer is sometimes used in hospital laundries.

Cylinders of washers may be one or two compartment by providing a partition in the center. Where a number of machines are installed, only two, perhaps, need be two-compartment type. Having compartments facilities keeping pieces separated according to size and classes. This kind of separation makes for more efficient handling at all stages of the process.

About 2 lb of dry laundry per cubic foot cylinder capacity may be figured for the washing process. Thus during one hour, in which one cycle is allowed, some 100 lb may be taken as the capacity of a 42-in. by 72-in. washer and 200 lb per hour may be estimated as the capacity of a high duty washer of the same size. These figures are for dry weight, and the cycle includes loading, washing, rinsing, emptying and unloading.

Washers are rated on the diameter and length of the cylinder inside. For instance, the cylinder of a 42-in. by 72-in. machine is 42 in. diameter and 72 in. long, inside measurements.

Flat-Work Ironers

Capacity of flat-work ironers is determined by width of machine, number of rolls, steam pressure carried and efficiency of venting and draining the rolls.

The proportions given in the instances cited indicate the approximate number required in a laundry. It is not advisable to install a flat-work ironer having less than five rolls, especially when blankets are to be laundered, which is usually the case in a flat-work plant. Where there is a smaller number of rolls, too much material has to be run through twice and sometimes, more. In general, a six-roll flat-work ironer is preferable. Care must be taken to see that the length of the rolls is well above the width of the widest sheets to be ironed.

Flat-work ironers are rated on length of roll in inches and number of heated rolls. Thus, a "110 five-roll" machine has five rolls, each of which is 110 in. long.

Tumbler Dryers

The proportion of tumbler dryers to washers is established in the instances given. For an unusually large percentage of blankets and similar articles, larger tumbler dryers would be required, and in a locality where few blankets are used, they may be smaller.

Tumbler dryers are high duty dryers for work that could be done in the dry rooms. Although not absolutely necessary, they are regarded as being practically indispensable nowadays.

Tumbler dryers are rated on diameter and length of cylinder for the reception of material to be dried. A 42-in. by 72-in. tumbler has a cylinder 42 in. in diameter and is 72 in. long.

Each of these machines is provided with a small hot-air fan as part of the machine with the outlet on the back, provision being always made for a duct the size of the air outlet to discharge out of doors. The lint often carried in this air makes its discharge into the rooms objectionable.

Water, Steam and Power Requirements

The minimum amount of water, steam and power required may be expected to be about 100 gal of cold water, 80 gal of hot water, 30 lb of steam and 1 kw of

Size of Washing Machine	Gal Hot Water per Hour per Washer	Capacity in Storage Tank per Washer
Small	150	100
Medium	200	150
Large	300	200

electricity per 100 lb of clothes (dry). Average laundries will exceed these figures by 50 per cent and small laundries by 100 per cent.

The instantaneous demands for steam and water are much higher than the average demand, and many installations have had to provide an increased supply of hot water and steam, after being placed in operation.

Due to the heavy instantaneous demands, especially in a small laundry, it is well, wherever practicable, to obtain the supply from the main boiler plant where these drafts will not be felt. In this case, the amounts given in the accompanying table are satisfactory for a medium size flat-work laundry having standard washers. If high duty washers are used, they should be doubled.

If an individual boiler is installed for the laundry, or if, for any other reason, the load curve must be flattened out, the use of larger tanks and consequently, more storage, is necessary. Tanks double the sizes given in this table are often found necessary.

The average drying tumbler, as one which is 42 in. by 72 in., requires about 600 lb of steam per hour. Steam requirements of other machines and dry room combined do not exceed one-third that of the drying tumbler. Although washers have steam connections for which considerable steam is used, a reduced amount of hot water is used. In some cases, steam is liable to carry oil in suspension; hence, its use in washers is objectionable in such cases.

The maximum demand for electric feeders should be taken at full connected load for direct current, and for alternating current, the power to operate the largest washer should be added to the full connected load. The latter requirement is intended to provide for reversal of machines which is negligible with direct current but considerable for alternating current.

Washers require from 2 to 8 hp for drive, according to size, plus 100 per cent for reversing when alternating current machines are used. Extractors require 2 to 5 hp to drive, depending upon size. Tumblers take from 3 to 6 hp to drive, flat-work ironers from ¾ to 2 hp and electric irons from 500 to 1000 watts.

Steam Pressure Required

The capacity required of the flat-work ironer largely fixes the steam pressure required, as all other machines may use steam at widely varying pressures without appreciably affecting their operation or capacity.

With a flat-work ironer of five or six rolls, 70 lb per sq in. is generally sufficient, although pressures of 90 lb are sometimes necessary in order to get the full capacity from the machine.

Sixty pound is ample for the disinfector and small presses.

Pipe Sizes and System of Piping

The branches to each machine must be large enough to meet instantaneous requirements but the steam and water mains need not exceed one-half the combined area of the branches, as more than two heavy machines are not likely to be drawing full capacity simultaneously.

Steam connections to washing machines are ¾ in. to 1 in., according to size; to tumblers 1 in. to 1¼ in.; to dry rooms, usually ¾ in.; to flat-work ironers ¾ in. to 1 in.; starch cookers ½ in. and to garment presses ½ in. Although there are several small connections to the disinfector, the main supply should be 1 in.

The main steam lines should be properly graded and drained. If run at laundry ceiling, as is usually the case, the vertical drops need not be dripped to the individual machines for the coils and traps will take care of the condensation without any damage.

Return connections are made from all equipment except washers. Each coil or heating unit should be tapped separately through a thermostatic trap into a common return line which is vented and graded downward to a vented receiving tank.

A ½-in. return connection and ½-in. thermostatic trap is ample for any coil or heating unit except the tumbler which should be ¾-in. return with ¾-in.

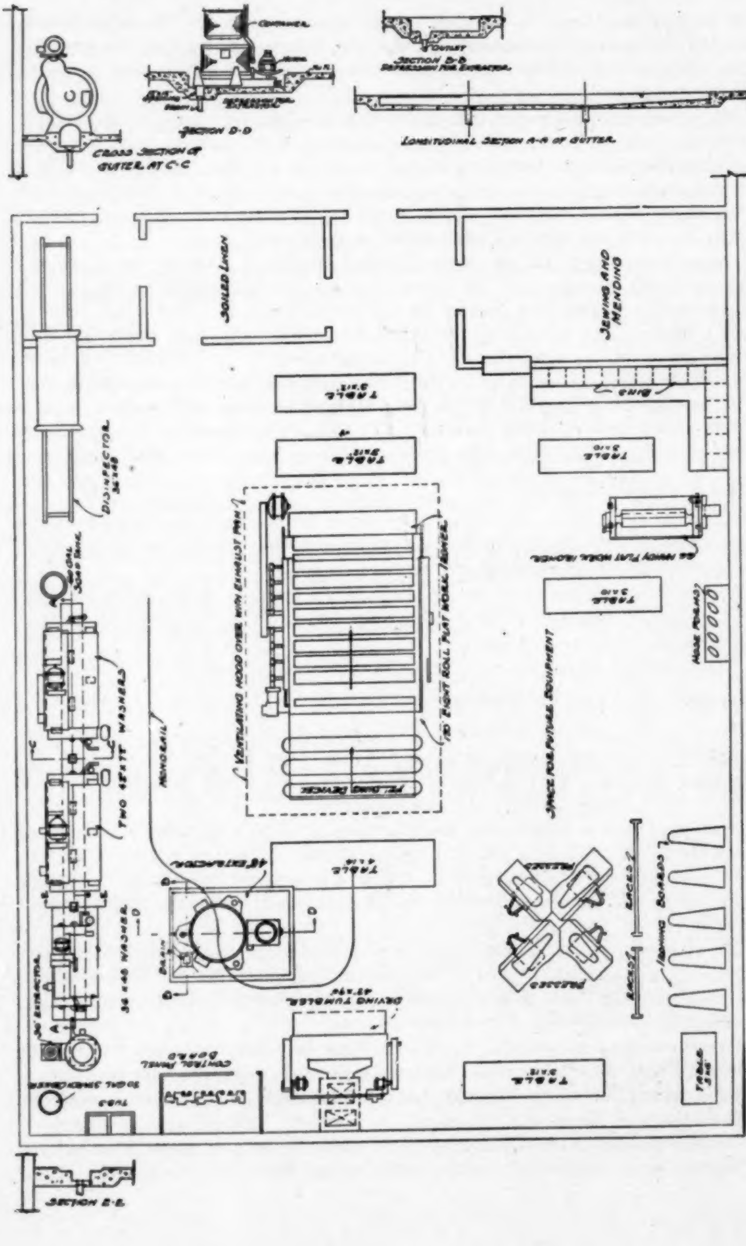

FIG. 1. ILLUSTRATES A LAYOUT OF A LAUNDRY FOR A 500-BED TUBERCULAR HOSPITAL

thermostatic trap. A ½-in. return and thermostatic trap will serve the smallest size drying tumbler, however, and return mains having an area one-half the combined area of the branches is ample, ½-in. being the minimum pipe size.

Although it is sometimes necessary to run return mains along the laundry ceiling, this is not advisable, for in such cases mechanical instead of thermostatic traps *must* be used, the condensation being lifted from floor to the return by the steam pressure. Such a main should be vented and run by gravity to a vented receiving tank.

Cold and hot water connections to washers are 1½ in. to 2 in.; ½ in. to soap tank and starch cooker and ¾ in. to each laundry tray.

High duty washers usually have two connections, each being the same size as for corresponding machines of similar dimensions. The extra capacity of these washers is dependent upon quick filling.

The individual control valves on the hot and cold water connections are placed in a convenient location for operation.

Water Softening

Soft water is required for satisfactory laundering, hard water forming insoluble compounds of calcium, magnesium and iron with the addition of soap, which cling to the fibres and spot the material. So much more soap is required when hard water is used, that it is desirable to have a water softener.

A number of such systems have been developed during recent years and have been found to be very satisfactory.

Layout of Equipment

For efficient operation, the arrangement of equipment must be given careful consideration, in order that it will fit in with the natural sequence of handling, from receiving to delivery.

Whenever possible, the laundry should be a separate building. The mistake has often been made of putting it on the second floor of the power house or in the basement of a dormitory. The ceilings of the building should be high, the rooms well ventilated by large windows and ample roof ventilators. Ventilating fans are often required.

All operations except those of receiving, mending, storing and delivery should be carried on in one room, as with this arrangement, better lighting and ventilation may be obtained and the general efficiency is increased.

Typical Layout

Fig. 1 illustrates a layout of a laundry for a 500-bed tubercular hospital.

In a 750-bed hospital for flat work only the following was sufficient.

One 36-in. by 42-in. by 84-in. disinfector.
One 36-in. by 48-in. standard washer.
Two 42-in. by 72-in. washers.
Two 60-gal soap tanks.
Two 30-in. extractors.
One 42-in. by 90-in. tumbler.
One 110-in. six-roll flat-work ironer.
Two 38-in. garment presses.
Two 49-in. garment presses.
One four-truck dry room, and one extra truck.
Five hand ironing boards.
Six 8-lb electric irons.
One 20-gal starch cooker.
One 36-in. by 60-in. starch table.
Four laundry trays.
Four wringers attached to trays.
Two 36-in. by 108-in. wood tables for use in connection with flat-work ironer.
Four 24-in. by 30-in. truck tubs.
One electric sewing machine.

Built-in tables and shelves are provided in the mending and delivery rooms for storage and sorting. The disinfector arrangement should make it possible to disinfect material before reaching the laundry proper. The parallel passage of flat work through washers, extractors, tumblers and flat-work ironer with that of the laundry trays, dry room and presses guards against any contact of washed and unwashed materials.

In Fig. 1, steam is furnished from a central plant, the hot water tank being placed in a room at one corner of the building, below the laundry-room floor. The ceiling height is 22 ft and 20 ventilators of the 36-in. size are provided in the roof. Mezzanine floors are to be found over the receiving and delivery sections in which toilet and locker rooms are located.

A hot water heating tank of 400-gal capacity, having a copper coil of 20 sq ft and operating on 65 lb of steam pressure takes care of the hot water supply.

The waste from the washers is conducted into a trench in the concrete floor, 30 in. wide, 4½ in. deep at each end and 6 in. deep in the middle. An 8-in. drain in the middle, protected by raised grating, leads to the sewer. In this way, most of the lint, which might cause trouble, is caught in the grating. It is important that washers do not discharge directly into a sewer on account of the stoppage resulting. A similar depression with drain is provided for extractor.

Water and steam distribution lines run overhead with individual drops to machines. Returns have individual thermostatic traps at each coil or unit requiring return and return mains are run below the concrete floors, encased in terra cotta pipes. The return mains are run by gravity back to a vented receiver in the power house some 500 ft distant.

Heating is done by vacuum system, with mains running at ceiling and returns in trenches below concrete floors in the same manner as returns from laundry equipment, then back to vacuum pumps in the power house.

Kitchen Equipment

The equipment considered here includes the usual cooking equipment found in the ordinary institution or hotel kitchen. Although there are other special appliances to be had, these, for the most part, require little space, steam and pipe connection and may easily be installed as an after-thought.

Character of Equipment

Nearly all so-called "steam" kitchen equipment, including steam tables, dishwashing machines, warmers and kettles may be obtained for either gas or steam heat.

For the small kitchen, where gas is available, the direct gas-fired equipment is preferable, perhaps, especially if a steam boiler must be installed. In a large kitchen, however, even if gas is to be used as fuel, it is better to install the usual steam equipment and a gas-fired steam boiler. In cases where employment of a licensed engineer is necessary, the use of direct gas-fired equipment might be desirable.

There are certain advantages which steam heated equipment has over gas in connection with steam tables, dishwashers and kettles. It can be left on a dry fixture indefinitely without damage, while with gas this is not true. The cost of upkeep and maintenance of steam equipment is considerably less, even in the event of there being a gas-fired steam boiler installed.

Although a range may be fired with wood, coal, oil or gas, the latter is to be preferred. The use of electricity for a range is practicable, but being more expensive, it is not so common as gas.

Proportioning Equipment

It is difficult to lay down general rules for proportioning equipment to be installed in the kitchen as so much depends upon the technique employed by the management. The proportions laid down below are subject to wide variation, on this account, and are based upon steam-heated equipment. No substantial difference in capacity exists, however, if gas-heated equipment is used.

Ranges

The smallest practicable gas or coal range has a top surface of about 7½ sq ft and an oven volume of about 4000 cu in., this size being considered sufficient

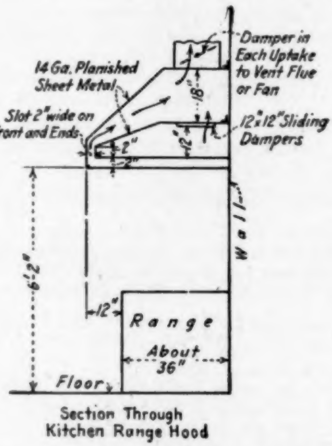

FIG. 2. DIAGRAM SHOWING A DOUBLE RANGE HOOD

for about 25 persons on the theory that no kettle or roasting ovens would be installed in such a small kitchen, that all cooking of that sort would be done on the range. About 15 sq ft top surface and 8000 cu in. oven space is ample for 100 persons, and for each additional 100 persons, 7½ sq ft top surface and 4000 cu in. oven space should be allowed.

In order to produce sufficient dependable ventilation, it is desirable to have a steel range hood with a fan to create a good draft. In most cases, the range stands against a wall, the hood projecting 12 in. or more beyond the edge of it at the ends and front. A double hood such as shown in Fig. 2 is generally conceded to be the best of this type of equipment. A slot 2 in. wide runs the full length of the hood on the side and in front. There should be sufficient volume of air to produce a velocity of 900 fpm through the slot, or 150 cu ft of air per minute per linear foot of same.

Cook's Table

Constructed of steel, the cook's table is about 3 ft wide and is generally placed opposite the range front, some 4 ft from it. There are several types of such tables, one of which is provided with drawers for knives and similar equipment. Another kind has warming closets in the base, while another type frequently found in cafeterias, is a combination of cook's table, warming closet and steam table.

The bain marie is generally built in the cook's table or installed at one end of it; a cook's sink, usually a simple galvanized affair minus a back and about 18 in. by 36 in. by 6 in., is often placed at the end of the cook's table, opposite the bain marie.

The sauce pan rack is almost invariably the full length of the cook's table and consists of two rails or racks about 24 in. apart, one rail over each edge of the table, with a semi-circular connecting rail at each end. Hooks attached to the rails provide storage room for the pans. Although small racks are often attached to the table, the larger ones are supported from the ceiling, thus leaving the table free.

Kettles, Ovens and Steamers

Stock kettles, roasting ovens, vegetable steamers and cereal cookers are seldom installed in kitchens serving less than 50 persons as such cooking is done on the range.

The following sizes of steam kettles and roasting ovens are suggested:

One 25- or 30-gal kettle and one 25- or 30-gal roaster to serve 250 to 300 persons.
One 40- or 50-gal kettle and one 40- or 50-gal roaster to serve 350 to 450 persons.
Two 30- or 40-gal kettles and two 30- or 40-gal roasters to serve 500 to 700 persons.
Three 30- or 40-gal kettles and three 30- or 40-gal roasters to serve 750 to 1000 persons.

In this way, the requirements of hotel kitchens would be:

100 rooms—One 40-gal steam kettle.
250 rooms—One 50-gal steam kettle.
One 25-gal steam kettle.
One 10-gal steam kettle without cover.
500 rooms—One 75-gal steam kettle.
Three 60-gal steam kettles.
One 20-gal steam kettle without cover.
800 rooms—Two 100-gal steam kettles.
Two 75-gal steam kettles.
One 60-gal steam kettle.
One 40-gal steam kettle, tilting type.
One 25-gal steam kettle without cover.
1000 rooms—Four 100-gal steam kettles.
Three 80-gal steam kettles.
One 60-gal steam kettle.
One 25-gal steam kettle without cover.
One 60-gal steam kettle, tilting type.

For a hospital or institution kitchen, where the staff dining hall and diet kitchens are served by the main kitchen, the following equipment is suggested:

50 beds—One 25-gal steam kettle.
150 beds—One 25-gal steam kettle.
One 40-gal steam kettle.
250 beds—Two 60-gal steam kettles
One 10-gal steam kettle without cover.

Vegetable steamers are not usually installed for less than 50 persons, such cooking being done on other equipment. A 2-bushel vegetable cooker will serve 200 persons, while an additional two-bushel steamer will be sufficient for 200 persons until three such steamers are installed, which would serve 1000 persons. Vegetable steamers are made in 1-, 2- and 4-bushel sizes.

The compartment type of cast-iron vegetable steamers is not used much nowadays on account of being difficult to keep clean, leakage between the sections and the impossibility of keeping the doors tightly closed.

Cereal Cookers

One 15 gal cereal cooker will serve 100 persons and an additional one for each 200 persons besides should be installed, until there are four such cookers; this number is sufficient for 1000 persons.

Coffee Urns

A pair of coffee urns, each having a capacity of 5 gal per 100 persons should be provided, urns having capacity of 3 gal each taken as the minimum size to be installed. Such urns are generally fitted with a water boiler between, which has a capacity equal to that of the two urns combined.

Steam Tables

In cafeteria kitchens, the steam tables tend to take the place of the cook's tables and are usually placed opposite the ranges and stock kettles. The sizes of such tables are usually as follows:

Five-foot long unit—	Two meats Four pots Two gravies	will serve 150 to 250 persons.
Six-foot long unit—	Two meats Four pots Two gravies	will serve 250 to 300 persons.
Seven-foot long unit—	Three meats Six pots Two gravies	will serve 300 to 500 persons.

Sinks

Pot sinks, work sinks, meat and vegetable sinks are usually of galvanized sheet iron, with 12-in. backs and drain boards at each end.

One 36 in. by 26 in. by 16 in., two compartment pot sink will be large enough for 50 persons; for larger kitchens, one which is 72 in. by 27 in. by 16 in., with three compartments.

Few kitchens would require a larger sink than the one mentioned.

A vegetable sink of corresponding size should be installed in the vegetable preparation room and a sink of the same size should be provided in the general work room.

Smaller sinks are used for other purposes, and these are generally 30 in. by 18 in. by 6 in.

Fish Refrigerator

Fish cannot be kept in an ordinary refrigerator, and it is best that they be packed in ice. For less than 100 persons, some makeshift may be used unless

fish is used often; for a large kitchen, a fish box should be provided, the smallest procurable having an internal capacity of 9 cu ft and sufficient for 200 persons. Installation of larger sizes would depend upon amount of fish to be in storage at one time.

Ice Cream and Sherbets

A 20-qt motor driven freezer will provide ice cream for less than 200 persons. For over this number, however, a 40-qt freezer should be used, and in connection with this process, it is well to consider the advisability of installing an ice-cracking machine. An ice chest for the kitchen is generally 22 in. by 28 in. by 33 in.

Mixing Machines

Mixing machines for batters, salad dressings and such preparations are found in most kitchens of this sort, and these may vary in sizes, ranging from the 15-qt to the 60-qt and 80-qt capacities. As the 60-qt size is sufficient for 300 persons, it is perhaps the most popular.

Vegetable Peelers

These should be in the vegetable preparation room. They vary in sizes from the 25-lb peeler, which serves 100 persons, to the 60-lb size, this being sufficient for 1000 persons. It is advisable to install a small sink having a 3-in. waste outlet adjacent to this machine, with a cold water faucet over it, in order to take care of the waste. The hopper should be close to the floor and this will require that the trap be placed at the ceiling below.

Dishwashing Machines

Dishwashing machines are rated by the number of dishes washed per hour; the rated capacity of the machine should be 10 times the number of meals to be served in any hour.

Such washers are built with one, two and three tanks and a like number of separate sprays. In this way, mixing of wash and rinse waters is avoided. The single tank machines are available in such small sizes, that they are suitable for diet kitchens and similar kitchens which are very small.

Of the three tank style, there are certain types which are semi-sterilizing in their operations because the last spray through which the dishes pass is a mixture of hot water and steam. Necessary in hospital kitchens, this type is not essential in the ordinary kind.

A dishwashing sink is about 30 in. by 18 in. by 6 in., with a wooden drain board at each end should be provided somewhere near the outlet end of the washer. It is unavoidable that a dish now and then passes through the average machine without getting properly washed, hence, must be picked up as it passes out.

Sterilizing dishwashers and possibly, some others, discharge the dishes sufficiently hot to dry themselves almost instantly, but where this is not the case, the dishes must be hand dried.

Broiler

A broiler may not be required in an institution kitchen but it is always necessary in a hospital, hotel or club kitchen, if less than 100 persons are to be served.

Broilers are obtainable for electric, charcoal or gas heat and in widths of 18 in. to 40 in., are approximately the same depth as ranges and are usually placed at one end so that the range hood can be utilized for carrying away the odors.

Miscellaneous

Meat blocks, meat slicers, coffee mills and bread slicers are usually found necessary, and in a large kitchen or bake shop, a barrel truck is frequently advisable.

Short Order Kitchens

These are often required in hospital kitchen and for one of ordinary size, a 24-in. by 30-in. gas or electric hot plate is generally used, a small steam heated warming closet, an 18-in. by 30-in. by 6-in. sink and a small work table.

Diet Kitchens

Diet kitchens are generally provided in hospitals, and these are distributed among the wards and bedrooms. In certain kinds of hospitals and in institutions the main dining rooms are built adjacent to the main kitchen and the "ambulant" patients and personnel go to these, but the diet kitchens are for the bed patients.

The bulk of the preparation for these kitchens is performed in the main kitchen, being carried from there in heated food carts.

Equipment for a diet kitchen usually consists of an 18-in. by 24-in. gas or electric hot plate, one four ft steam table with meat pan, four vegetable jars and two gravies; a warming closet below the steam table, a 1500-piece capacity dishwashing machine, a small refrigerator and an 18-in. by 30-in. by 6-in. sink.

Bake Shops

If the baking of bread and pastry is to be done on the premises, the bake shop should be apart from the main kitchen, although adjacent to it. In a contagious hospital, however, where the main kitchen may be in a location subject to infection, the bake shop should be well away from it.

Ovens are rated on the capacity at one baking, so that installed in the bake shop there should be one-half of one pound loaf per person. In larger institutions, the oven may be of smaller relative capacity, but based on more continuous baking operations.

A pastry oven having an aggregate baking surface of some 5 sq ft per 100 persons should be provided in the bakery. In some cases, bread is purchased, and the oven is used for pastry only, thus eliminating the necessity of having the separate bake shop, and installing the pastry oven in the kitchen.

If gas is not available for baking or pastry ovens, electricity is the next best type of fuel. Although coal and wood are sometimes used, they are not entirely satisfactory for baking. Of all types of cooking equipment, ovens have proved the most adaptable to electric heating.

In bake shops generally, a bakery combination consisting of sifter, elevator, tempering tank, dough and cake mixers is considered necessary. Such outfits are rated on the capacity of dough mixer in barrels per hour and sizes from one to four barrels are available in this combination. One-half barrel is rated per 100 persons in a small institution and one-quarter barrel for larger hospitals.

Bowls of various sizes from 30- to 80-qt capacity are attachments to the cake mixing part of the combination, in which sponge beaters, batter beaters, dough

beaters are operated; this part of the bakery combination is practically the same as the mixing machine installed in the main kitchen. With the bake shop adjacent to the main kitchen, one mixing machine outfit may serve both places; in any event, the bowls and attachments should be interchangeable.

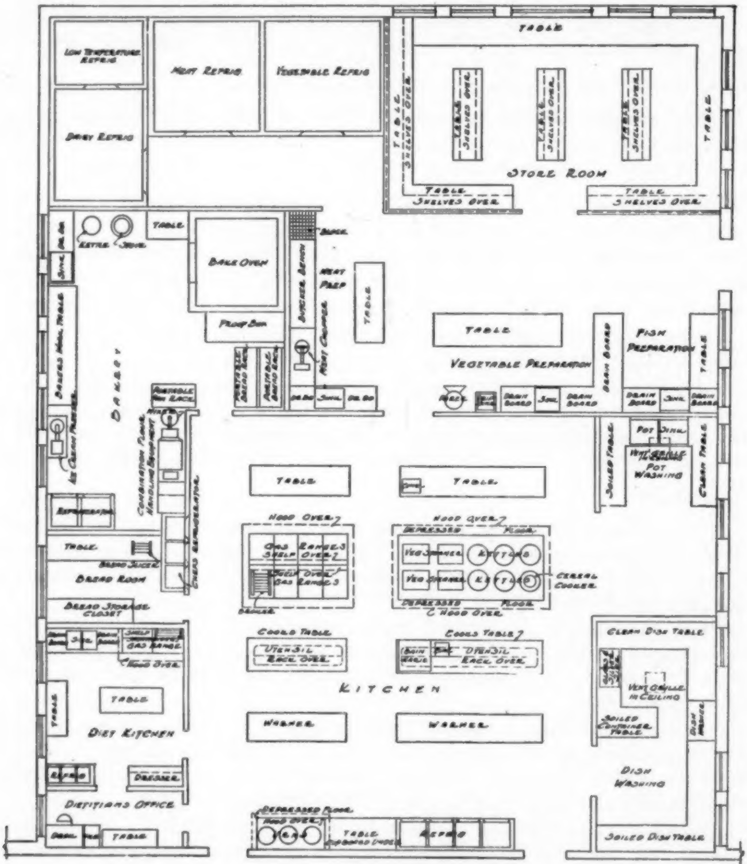

FIG. 3. SHOWS KITCHEN FOR 500-BED HOSPITAL

A steam heated or gas heated proofing oven having a capacity of 12 sq ft shelf space per 100 persons should be provided in connection with the bread oven.

A dough trough on rollers, from 4 ft to 10 ft in length is necessary in the bake shop. The work table is usually larger than the dough trough, with space underneath for storing the trough. A pair of platform scales of about 400-lb capacity are needed in a bakery, and in a good sized kitchen, a similar pair should be provided.

Cafeteria Restaurant Service

Many institutions have cafeteria restaurant service, and the coffee urns, milk urns and ice cream cabinets are placed behind the rail. Provision must be made for steam tables, cold pans and salads, also. The cold pans are 24 in. wide, are available in various lengths and refrigerated by ice or coils. Most of such pans are some 4 ft long. A small sink should be placed behind the cafeteria rail.

Refrigerators

Where a central cooling plant is used, as is often the case, built-in refrigerators should be installed. Portable refrigerators, if used and connected to the central plant, will soon result in a sloppy installation of refrigerant piping due to temptation to move them frequently.

Electric, self contained, portable refrigerators are advisable where no central plant is provided.

An ice-cooled refrigerator is the last resort, except in hospital diet kitchens where the desirability of having a small amount of ice on hand makes a small, ice-cooled refrigerator suitable.

In the main refrigerators there should be separate compartments, or separate refrigerators, for the storing of meats, vegetables, dairy products and salads and one, also, for miscellaneous products.

Garbage refrigerators are often necessary in congested urban communities, and these should hold two garbage cans for every 70 meals served. The destruction of garbage in incinerators in such communities often gives rise to complaints on account of the odors.

Layouts

The sequence of operations should be kept in mind in the planning of a kitchen. Every part of the room should be accessible in order that the service may be continuous, with as few steps as possible being taken by waiters and cooks. Cross travel or counter rotation should be avoided. Although enough space is desirable, this factor can be over-emphasized. Possible future expansion should be taken into account and the layout should be so planned that the minimum amount of help will be required. Figs. 3 and 4 are kitchens for a hospital and hotel, respectively.

The kitchen should be near enough to the dining room to insure quick service. In general, exhaust ventilating fans should be installed in the kitchen rather than in the dining room; in this way, the draft will be from dining room to kitchen when the communicating doors are opened rather than the other way round. The placing of service pantries between kitchen and dining room help to maintain this condition and also shut out noises issuing from the kitchen.

Adequate space should be provided for receiving and storing supplies and there should be but one entrance from the outside to the kitchen.

The preparation spaces, pantry and ice cream shop should be as close as possible to the service counters of the kitchen. Adjoining the dishwashing room should be a room for soiled linen; clean linen rooms should be placed close to the service pantries. Storage space must be had, also, for surplus dishes, and this should be near the kitchen.

Ventilation requirements vary greatly, depending upon opportunities for

natural ventilation. It is only in rare instances, however, that natural ventilation which will be sufficient for a busy kitchen may be obtained. When a fan is needed, it should change the air each 3 to 5 min in kitchen and pantries. The ventilation of the range and of the dishwashing room should be separate from that of the rest of the room so that these units may be operated when others can be shut down.

Work tables and china closets must be provided, also, as well as tables for soiled and clean dishes, the latter of which are usually built of wood, 2 in. thick and covered with galvanized sheet metal.

The cereal cookers, steam cookers and soup kettles are generally grouped and the floor in this place should be depressed about 1 in. and provided with a drain for waste.

A hood is usually desirable over these cookers, similar to the one over the range, and the ventilation through this hood may be considered as part of the kitchen ventilation.

The following equipment was satisfactory for a 500-bed contagious hospital kitchen with dining room for 350 ambulant patients:

One oil burning range 30 ft long.
One 20 ampere hot plate for short orders.
One 20 ampere grill.
Two 60-gal stock kettles.
Two 2-bu vegetable steamers.
Two 100 lb meat roasting ovens, steam.
Two 25-gal cereal cookers.
One sterilizing dishwasher, 5000 pieces an hour.
One meat chopper.
One cook's table 18 ft long.
One 36-in. by 36-in. bain marie sink.
One 24-in. by 50-in. roll warmer.
One 24-in. by 48-in. by 60-in. plate warmer.
One coffee urn set consisting of two 50-gal urns, one 100-gal water urn, with stand and cup warming closet.
One 24-in. by 60-in. cold pan.
One ice cream cabinet with three 5-gal containers.
One 72-in. meat table.
One 12-ft vegetable table
One 18-ft double sauce pan rack.
One 40-lb potato peeler.
One 80-qt mixing machine.
One steam table with warming closet, 10-ft long having four meat pans, three gravy boats and four 2-gal vegetable jars.
One motor driven drink mixer.
One 40-qt motor driven ice cream freezer, with provision for storage of ice cream in refrigerators.
One 48-in. by 24-in. by 14-in. pot sink, two compartment.
One 36-in. by 24-in. by 14-in. pan sink.
One 60-in. by 20-in. by 12-in. two compartment vegetable sink.
One 48-in. by 24-in. by 12-in. two compartment vegetable sink.
One 30-in. by 20-in. by 6-in. meat sink.
One 16-in. by 16-in. by 12-in. sink for potato peeler waste.
One 30-in. by 20-in. by 6-in. sink for short order kitchen.
One 30-in. by 20-in. by 6-in. sink for service counter.
Two drinking fountains with cup filling faucets.
One 22-in. by 28-in. by 33-in. ice chest.

The bake shop equipment consisted of the following:

One 250 loaf electric bake oven.
One pastry oven having 25 sq ft. baking surface, electric.
One 2-barrel bakers combination with 3-barrel hopper and mixing machine attachments with 80-qt bowl.
One 8-ft dough work table.
One 6-ft dough trough.
One proofing oven, shelf area 60 sq ft, electric.

Water Requirements

Hot water requirements for kitchens may be figured on the basis that each sink faucet will take 20 gal per hour and a 1000-piece dishwasher will take 40 gal per hour. The latter is somewhat under the control of the operator, for when more steam is used, less hot water is necessary. Although cold water requirements

are about double those of hot water, all that is needed is to have pipes of large enough size.

Steam Requirements

The dishwasher will consume from 50 to 100 lb of steam per 1000 pieces per hour. An allowance of 1½ lb steam per person per hour will meet all steam requirements for the kitchen except those of dishwashing machine and water heating. A 25- to 30-hp boiler will take care of the average steam needs for a kitchen serving 300 to 500 persons. The annual steam consumption of such a kitchen will be about 2,000,000 lb.

Power Requirements

Dishwashers require 1 to 2 hp per 5000 pieces per hour, with the smallest machine using ½ hp motor. Kitchen mixing machines take ½ to 1 hp, bake shop combinations 2 hp per barrel rating, 17 in. meat, food and vegetable choppers 1 hp and the 20 in. size 2 hp. The horse power for coffee grinders, knife polishers and silver burnishers approximate one-fourth. Vegetable peelers take from ¼ to 1 hp according to the size. One-half horse power may be allowed for a 25-qt ice cream freezer and 1½ for the 40-qt size.

Electric Cooking Requirements

In a complete electric kitchen having ranges, broilers and hot plates run by electricity, the number of watts required for a complete meal will average as follows, this being the minimum:

General restaurant—400 to 500 watts.
Institution kitchen—175 to 200 watts.
Colleges, boarding schools—250 to 350 watts.
Clubs—450 to 600 watts.

Unless there is care used in operating electrical equipment, there is apt to be considerable waste.

Steam Pressures

Steam cooking requires not less than 30 lb and not over 50 lb gage. Pressures as low as 1 lb, however, are ample for some appliances such as food warmers. All jacketed pots have safety valves which are usually set at 50 lb; usually vacuum breakers are required on jacketed fixtures for the prevention of collapse under vacuum conditions.

Pipe Sizes

Hot and cold water connections to sinks are ¾ in.; hot water only is provided for the dishwasher in pipes varying from 1¼ in. in the largest size and ½ in. in the smallest. One-half inch connections are used for hot and cold water to the bain marie and ½ in. ones for coffee urns. Stocks kettles, vegetable steamers, soup kettles and potato peeler sinks are often provided with a ½-in. faucet over each, supported independently of the fixture. It is generally only hot water, however, that is so connected. Drinking fountains and water coolers require ½ in. connections.

In determining the size of mains to care for two or more items, it is usually satisfactory to make the mains one-half the sum of the areas of the branches.

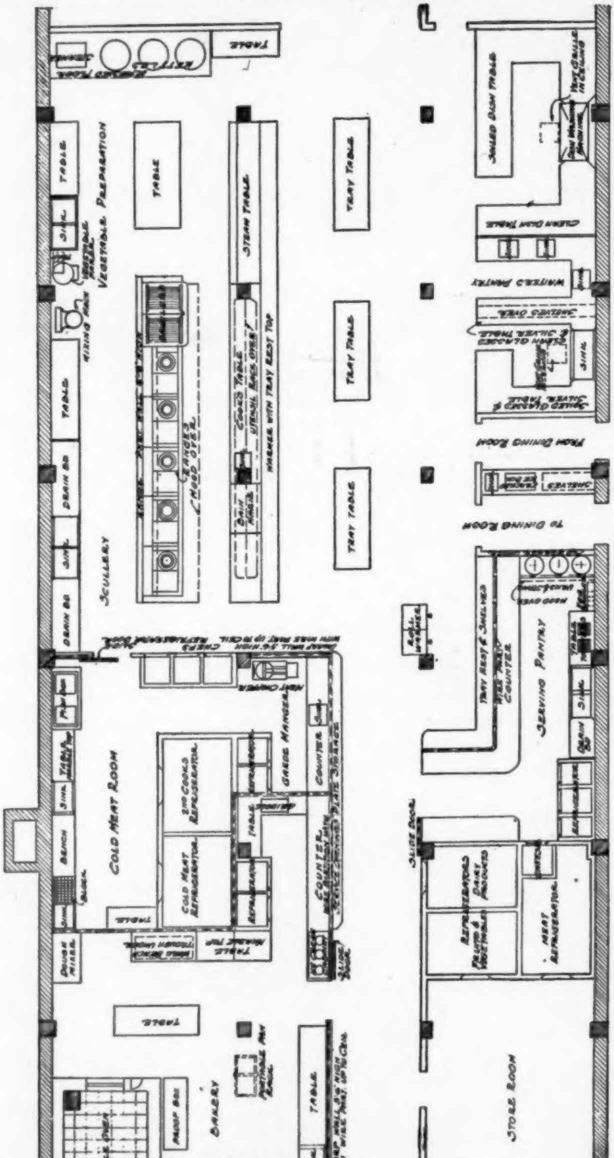

FIG. 4. KITCHEN FOR 500-ROOM HOTEL.

Steam Connections

Steam connections to jacketed kettles, soup kettles, vegetable steamers steam tables, coffee urns, plate warmers and roll warmers are ½ in. for smaller and ¾ in. for larger sizes while for dishwashers, steam connections are from ½ in to 1¼ in.

In determining the size of mains required, it is advisable to make the area of two or more items about one-half the sum of the branch areas, keeping the sizes well up toward the ends of the runs.

Return Connections

Return connections are uniformly ½ in. in size and are required for all equipment having steam connections except the spray to dishwashers. The coil in this machine, however, requires a return connection.

Each unit or separate coil should be provided with a ½-in. thermostatic trap. If more than one unit or coil is connected to the same trap, air binding will result. Connections of the same size are made to return mains.

Wherever it is possible, the return mains should run below the level of the equipment, and should be vented by extending them up to atmosphere at suitable points, being run by gravity to an open receiving tank.

If necessary the return mains may run above the equipment, and in such an event, mechanical traps the same size as thermostatic traps will have to be used instead of thermostatic traps, and the condensation lifted to the return by the steam pressure in the line. The return main should be vented, also, and run to an open receiving tank just as if thermostatic traps were used.

Drainage of Steam Mains

The steam mains should be graded and properly drained in manner similar to any other steam piping system. It is not necessary, ordinarily, to drip the drops to the various pieces of equipment, as the jackets, coils and thermostatic traps will take care of a limited amount of condensation when opened, without any damage. Steam branch connections running below equipment will generally drain themselves back into the main.

Gas Connections

It is very important that gas connections of ample size be provided to all equipment using gas. Although the effect of a steam connection one or two sizes too small might not be noticeable, such reduction of gas connections would cause serious complaint.

The information on the amount of gas consumed per hour with the different fixtures should be obtained from the manufacturer, and the gas piping should be sized accordingly. Some excellent tables for this purpose have been furnished by the *American Gas Association* and are of particular value when the gas consumption and length of run are known.

The following is given as an approximation of connection sizes usually found, these being for equipment of average size:

Pastry oven	¾ in.	Ranges	1 to 2 in.
Bake shop oven	1 in.	Dishwashers	¾ to 1½ in.
Broilers	1 to 1¼ in.		

With range separate connections to top and ovens of ¾ to 2 in. size are sometimes used, and in some cases, two or more connections to top and ovens both are used in order to equalize pressure. In general, the equalized area, of a 1 in. pipe for each 3 ft length of range will approximate the total requirements for both top and ovens. Tables of equalization of pipe areas may be found in almost any engineering handbook. For small ranges, such as in diet kitchens, 1 in. is usually sufficient. Salamanders and warmers usually take ¾ in. and steam tables 1 in. connections.

In any case, whether the approximate sizes given for connections, are used or not, the main must be sized in accordance with tables given by the *American Gas Association* and various handbooks available, which take both the load and length of run into account.

Waste Connections

Such connections from fixtures are usually as follows:

Sinks	2 in.	Bain maries	1¼ in.
Dishwashers	2 in.	Drinking fountains	1¼ in.
Vegetable peeler sinks	3 in.	Refrigerator drip pans	1¼ in.
Steam tables	1¼ in.		

Vents

The steam-jacketed stock kettles, soup kettles and vegetable steamers are generally constructed with 3 in. vents in the tops for the cooking chambers. These should be connected together into one 3-in. main vent and run into the vent from the range hood. It is advisable to use standard weight galvanized pipe, for although sheet metal pipe may be employed, it fails very quickly when used in such a location.

Grease Traps

The dishwasher, pot, pan and any type of sinks into which grease may find its way, should be provided with grease traps. When these are used on individual fixtures, the cold water supply to the fixture is connected to pass through the jacket of the grease trap. In placing such traps, it is a common failure to put them in such a position that it is difficult to remove the covers for cleaning, which should be avoided. Individual grease traps are generally made of cast iron.

Institution kitchens, should, where possible, be provided with a main grease trap on the main waste line from these fixtures outside the building. This arrangement facilitates cleaning and is much more efficient as a grease extractor than the individual fixture grease traps. There is no objection to leading wastes from vegetable sinks which might not otherwise have grease traps, through this trap, if it is more convenient to connect them into the common waste line. Wastes, however, from sanitary fixtures or those passing considerable volumes of cold water, should not be connected through this trap. Sanitary wastes are obviously objectionable and too much cold water will cause the grease to coagulate before reaching the trap.

In all cases, it is advisable to make the waste lines to the trap a little larger than would otherwise be the case in order to allow for a certain amount of coagulation and to provide an ample number of cleanouts for removal of stoppage.

The main cold water supply line to the entire kitchen should be led through coils of pipe in the trap, and the sewer should be by-passed around the trap with valves which will make it more convenient to clean the trap.

For a kitchen feeding 500 persons, such a trap 6 ft by 6 ft by 4 ft below the invert of the sewer is sufficient. This is simply a concrete box extended full size up to grade and provided with removable covers the full area of box. The outlet sewer should be turned down, inside the trap, to near the bottom of the trap, so as to gather the cooler water, and the inlet should be at the surface.

Two coils of 2-in. pipe, made up with return bends on 4½-in. centers, should be across the trap or box in order that the wastes may flow between the pipes. Coils should be placed 2 ft apart and 2 ft from each end and the pipes should be vertical, so that the grease may be removed easily. The two coils may have water connections either in multiple or in series, preferably the latter if 2-in. pipe is large enough to carry the entire supply. If in a series, the water should first enter the coil toward the outlet end of the trap.

Hospital Equipment

Every hospital must sterilize all instruments and utensils used in operations and treatments as well as all dressings before being used, bedpans and baby's milk and bottles. There must also be provision for the sterilizing of mattresses and dishes. All of which makes it necessary that a plentiful supply of sterilized water and distilled water be at hand at all times. In hospitals where mental and nervous diseases are treated, there must be hydrotherapy and electrotherapy equipment, also.

Instrument, dressing, utensil, dish and mattress sterilizers are usually rated by their internal dimensions in inches; water sterilizers by the gallon capacity of each of the two reservoirs, water stills by their capacity in gallons and pasteurizers as well as bottle sterilizers by their capacity in 8-oz bottles.

Selection of Sterilizers

Sterilizers and water stills may be had which are heated by steam, electricity, gas, kerosene or gasoline, the preference being in the order named. Steam is the only method, however, for which the entire line of sterilizers are ordinarily made. Such sterilizers call for less equipment liable to derangement, are safer and quicker in operation and more economical. Electric sterilizers most frequently used, are confined to the smaller sizes, while those heated by gas, kerosene or gasoline are used only as a last resort. Steam heating, in fact, is the only way in which it is practicable to sterilize large mattresses.

A steam heated sterilizer may operate dry indefinitely without any injury resulting, but this is not the case when other systems are used. Those which are electric should have equipment in order to protect them against low water and over pressure. When heated by gas, kerosene or gasoline, there should, at times, be devices for protection against pressure but not against low water. Difficulty often arises on account of the derangement to which such devices are always liable.

Electricity is superior to any other method only in the case of the solution warmer, where, due to its adaptability to automatic control, it is preferable.

Cost of Operation

The cost of operation of steam sterilizer equipment is very little more than the cost of keeping steam constantly available and the amount of use has little effect on the total cost of operation.

Electric sterilizers cost more to operate, depending upon the cost of "standby" service for them. This is obviously true, considering that the smallest instrument sterilizer is rated at 2200 watts and the largest dressing sterilizer at 12,000 watts.

Although costing more than steam, the gas and other substituted methods cost less than electricity.

Open and Closed Sterilizers

In the open type of sterilizer, the articles are immersed in a water bath heated by steam coils, the limited temperature being 212 F. With the closed type, there is no water bath and the articles are brought into temperatures corresponding to steam at 40 to 60 lb pressure per square inch.

The former type, although in use for a longer time, has the disadvantage of depositing lime or other impurities from the water on instruments, sometimes requiring them to be dried before storing.

Better sterilization is provided by the latter and newer type, and articles are dry when removed from it. Sterilizers of this kind must be used for dressings and fabrics.

Built-in Sterilizers

There is a growing tendency to use the built-in sterilizers in operating suites and other places where they are installed at one point. Only pressure type sterilizers which are loaded and unloaded from one end are adaptable to this arrangement. With this method, a thin partition is built flush with the sterilizer fronts with only the doors, gages and operating valves visible in the rooms, the bodies projecting into an unfinished space behind the partition, the latter being accessible for repairs. Thus, a better appearance is maintained, the sterilizer is more easily kept clean by the elimination of much exposed piping and the operator's attention is less likely to be distracted.

Although such sterilizers are less expensive in themselves, the cost of partition makes the total cost of built-in sterilizers more than for other types, although this is not prohibitive.

Proportioning Equipment

Requirements in different hospitals vary so widely that it is difficult to state just what number and size of sterilizers should be installed. The suggestions given herein are for what is considered ordinary conditions.

One 16-in. by 24-in. dressing sterilizer will care for two major operating rooms, possibly three, but for more than two such rooms, it is better to provide a 16-in. by 36-in. size or, to install two 16-in. by 24-in. sterilizers. When two such sets are used, there is provision for breakdown service, and one of them may be used as a pressure utensil sterilizer, in some cases, doing away with the necessity of installing a separate sterilizer for utensils.

One 20-in. by 20-in. by 24-in. open type utensil sterilizer will care for two major operating rooms unless the work is very heavy, in which case, one such sterilizer should be provided for each room.

A 9-in. by 10-in. by 20-in. open type instrument sterilizer should be provided for each major operating room, although one 10-in. by 12-in. by 22-in. may serve two such rooms. The 12-in. by 16-in. by 24-in. instrument sterilizer finds its use where it is convenient to sterilize from three or more major operating rooms at

one place. A 14-in. by 22-in. pressure type instrument sterilizer will easily serve two major operating rooms.

A small hospital might install one 20-in. by 28-in. pressure type dressing sterilizer which would serve both for dressings and utensils, although no breakdown service would be provided.

Water sterilizers usually consist of twin reservoirs of the same capacity mounted on one frame, one being for water kept hot by steam coil in the reservoir, and the other cooled by circulating cold water through a coil in the reservoir. Both, of course, sterilize their contents.

Water stills are part of the water sterilizing unit, consisting of still and reservoir, generally being mounted between the two water reservoirs.

Water sterilizers are available in sizes from 6- to 100-gal capacity of each reservoir and stills in capacities of 1-, 3- and 6-gal reservoir capacity. The total capacity of the two reservoirs in the water sterilizer should range between 5 and 10 gal per major operation per day.

The 3-gal still would suffice for hospitals up to 100 beds and the 6-gal size should be used for larger ones in connection with the water sterilizers.

Stills of the same capacity are obtainable which are separate from the water sterilizers, and they are frequently needed in the maternity department.

The dressing, instrument and utensil sterilizers are primarily for the major operating rooms, and should be in or adjacent to them. Where the operating rooms are scattered, all sterilizers with the exception of that used for dressings, should be duplicated for each location. Where built-in sterilizers are used, the fronts may open directly into the operating rooms, or when a single sterilizer installation is used for more than one operating room, it should be located in a separate room between them. Uncontrolled sources of heat, however small, should be avoided as much as possible in an operating room. In hot countries this is very important, for oscillating fans are usually not allowable in such rooms. In each minor operating room, such as for eye, ear, nose and throat, and in the dental clinic, a small electric sterilizer usually about 5 in. by 6 in. by 16 in. for instruments should be provided. These are made with brackets for wall mounting.

In the surgery and maternity departments and at convenient locations to rooms and wards, blanket and solution warmers should be placed; these being available in sizes about 18 in. by 24 in. by 72 in. and 18 in. by 30 in. by 72 in., simply heated and divided into compartments. For wards and rooms they should be placed in or near the utility rooms.

A surgical dressing room may use the equipment of the main sterilizing room if adjacent to it, but if this is not possible, separate equipment should be provided. A 6-gal water sterilizer, one 8-in. by 9-in. by 18-in. instrument sterilizer and a 16-in. by 15-in. by 20-in. utensil sterilizer will usually suffice for a small hospital.

In the laboratory there should be installed an autoclave, which is a dressing sterilizer without the jacket, for the wet sterilizing of material. A 16-in. by 24-in. size will be large enough for a 100-bed hospital, although two larger sizes are available which are 18 in. by 26 in. and 22in. by 30 in.

Utility Rooms

These should be provided on each floor and should contain, in addition to combination sink and tray, one 20-in. by 24-in. utensil sterilizer and one bed-pan

sterilizer. Utility rooms serving surgical and maternity patients should also contain an 8-in. by 9-in. by 18-in. instrument sterilizer and a 10-to 15-gal water sterilizer.

Nursery

Equipment for pasteurizing milk and sterilizing bottles must be provided in the nursery. These may be combined in a small hospital, but in a large one, this method is not satisfactory. Pasteurizers can be had in 54-, 72-, 144- and 288-bottle capacity, 8-oz nursing bottles. About one hour is required for a cycle, as laws frequently require that the pasteurizing process consume 30 min. When bottle sterilizing is done separately, a pressure type sterilizer for nursing bottles is provided.

Dish Sterilization

Where there are contagious diseases, dishes may be sterilized in special dish-washing machines or, on a small scale, in a 20-in. by 20-in. by 24-in. utensil sterilizer.

Sterilization of Mattresses

Every hospital should be provided with a mattress sterilized. Although the smaller ones may be purchased which are arranged for gas or electric heating, mattress sterilizers are practicable only when heated with steam. The large size is 36 in. by 48 in. by 84 in., is the most widely used, and may be charged at one end and discharged at the other, thus offering the advantage of placing it in a partition between two rooms and effecting a complete separation of sterilized and unsterilized material.

Specifying Sterilizers

Different manufacturers have sizes of sterilizers which are approximately the same but not absolutely standardized as to sizes. Hence, it is advisable to give such limiting inside dimensions as may apply and minimum capacity in cubic inches when specifying sizes.

Piping for Sterilizers

Steam and return pipe sizes for sterilizers are almost nominal, the amount of steam used, being practically negligible when drawn from a central plant for heating. A ½-in. supply and return connection is ample for any utensil instrument or dressing sterilizer. Water sterilizers up to 15 gal require ¾-in. supply and ½-in. return; above that size, 1-in. supply and ½-in. return. For large mattress sterilizers, 1¼-in. supply and ¾-in. return connections may be used, while water stills up to 6 gal use ½-in. supply and return pipes.

When sizing the steam and return mains, it may be assumed that all sterilizers will be in use at one time in a small hospital and half of them in one which is larger. The mains can then be sized by the usual table of equalization of pipe sizes found in any handbook. A return main one-half the diameter of the steam main, with ½-in. as minimum size may be used.

It is a uniform practice to provide a thermostatic trap on the return from each coil or unit of the same size given above for return connection. The return main should be vented and condensation should return by gravity to a vented receiving

tank. With the mattress sterilizer located in the basement, a gravity return may be impracticable, hence a mechanical trap should be used in such cases, so that condensation is lifted to the vented tank with the steam pressure on the sterilizer.

Steam mains and risers should be dripped and graded in the usual manner. Generally all sterilizing equipment, with the exception of the mattress sterilizers, that may be in the basement, have steam and return connections from the floor or wall, preferably the latter.

When gas is used, a ½-in. connection is made to each instrument, dressing and utensil sterilizer. Water sterilizers up to 15-gal capacity should have ¾-in. gas line and above that size, 1-in. line. Mains should be sized on an equalization basis as explained for steam supply lines, assuming all sterilizers in use in a small hospital at the same time and half of them in a large hospital.

Waste and Vent Systems

All open type sterilizers must have waste connections similar to plumbing fixtures. Traps on these waste lines must be provided in the case of each kind of sterilizer except that of the bed-pan. The waste connections are seldom more than ¾ in. at the fixture, although they are 3 in. for bed-pan sterilizers. Such waste should be connected to the drainage system.

Vent connections are also necessary on such sterilizers in order to do away with steam and vapor. These usually take the form of a combined overflow and vent, connected into the rear of the chamber near the top. This connection is usually carried as one horizontal pipe to a suitable location, as close to the sterilizer as possible, where the vent is taken upward as directly as possible to atmosphere, and the overflow and condensation wastes are taken from the bottom through a trap, then connected to the waste lines of the plumbing system or connected to the waste from the body of the sterilizer between trap and sterilizer.

The usual sizes of combination overflow and vent connections to open sterilizers are as follows: utensil 1¼-in. up to 16 in. by 15-in. by 20-in. size and 2 in. for larger sizes; instrument 1¼ in. for all sizes; milk pasteurizers and sterilizers 1¼ in. up to 72-bottle size and 2 in. for those larger; bed-pan sterilizers 2 in. Vents to atmosphere are the same size as combination overflow and vent.

Where practicable, it is well to run the vents from sterilizers through the roof separate from other plumbing vent lines and to make them of extra heavy pipe, for the large amount of condensation in them often destroys standard weight piping in the smaller sizes rather quickly. A 2-in. vent line will carry half dozen sterilizers, but in cold climates, it should be at least 4 in. through the roof in order to prevent frosting over.

Care must be taken that the bases of vent lines from such equipment are drained through a trap into the waste lines of the plumbing system so that there will be no draining back into the sterilizers.

Closed or pressure type sterilizers require a vent to atmosphere which is connected to the chamber and provided with a valve. The relief valve, with which every closed sterilizer must be provided, is usually connected to this vent line on the line side of the valve. These vents are usually ¾ in. from each sterilizer.

Frequently, both open and closed sterilizers of all types, are provided with condenser exhausts, consisting of a special ejector, manipulated by a valve and throwing a stream of cold water into the ejector. The condenser exhaust is a part

TABLE 1—STERILIZER DATA

SIZE IN IN.	INTERNAL CAPACITY, CU IN.	POUNDS OF STEAM PER STERILIZATION	WATTS REQUIRED IN WIRING
	Cylindrical dressing sterilizers		
10 x 20	1,571	10	4,000
12 x 20	2,260	12	6,000
14 x 22	3,388	14	6,000
16 x 24	4,824	18	6,000
16 x 36	7,236	24	6,000
16 x 48	9,648	31	12,000
16 x 60	12,060	39	12,000
20 x 28	8,792	30	12,000
20 x 36	11,304	37	12,000
20 x 48	15,072	47	12,000
	Rectangular dressing sterilizers		
24 x 24 x 36	20,736 Approx.	55	Not standard
24 x 24 x 48	27,648 Approx.	70	Not standard
24 x 24 x 60	34,560 Approx.	88	Not standard
	Open type utensil sterilizers		
16 x 15 x 20	4,800	30	4,400
20 x 20 x 24	9,600	50	6,600
	Open type instrument sterilizers		
8 x 9 x 18	1,296	6	2,200
9 x 10 x 20	1,800	8	2,200
10 x 12 x 22	2,640	10	4,400
12 x 16 x 24	4,608	16	4,400
Water sterilizers. (Steam and watts are for each reservoir.)			
6 gal		15	3,000
10 gal		25	6,000
15 gal		37½	6,000
20 gal		50	12,000
25 gal		62½	12,000
35 gal.		87½	18,000
50 gal		125	24,000

of the fixture, when used, and generally wastes over an open funnel leading to the plumbing system waste through a suitable trap or connected to waste pipe from sterilizer chamber between chamber and trap. When separately connected, a 1¼-in. trap should be used. The purpose of the open funnel is to prevent the creation of a pressure on the plumbing system.

A ½-in. water supply connection should be made to each sterilizer provided with this ejector, and will carry a large number of such ejectors for the operation requires only a short time and it is seldom that more than one sterilizer will be ejecting at the same time. Any pressure above 25 lb per sq in. will operate these ejectors, the higher pressure doing the most efficient work and requiring less water. Cold water only is used.

Local and state plumbing codes generally have certain requirements regarding the piping of over-flow, waste and vent connections from sterilizers into the plumbing system. Where such codes apply, their requirements should be investigated and the work designed accordingly.

A furred space in the walls behind the fixtures for the pipes and connections should usually be provided on account of there being so many connections to

sterilizer equipment. Although connections may be made to the floor, for sanitary reasons it is better to make them to the wall. Built-in sterilizers, of course, avoid the necessity of having the false partition. In order to facilitate their cleaning, it is often advisable to place the traps at the bottom of vents, at basement ceiling to make them accessible.

Sterilizer Data

Data on sterilizers of various types and sizes are given in the accompanying Table 1, although it must be remembered that the standards of different manufacturers, dimensions of sterilizers and capacity in cubic inches vary. In layouts for electric sterilizers, the actual requirements must always be investigated, otherwise, much difficulty might result.

Hydrotherapy Equipment

This type of equipment is generally considered essential in psychiatric hospitals and consists of one set of fixtures unless the hospital is very large. In the set are included needle shower and rose spray, perineal stool and sitz bath with liver spray and one control table. The maximum momentary use of water for outfit amounts to about 40 gal of cold water and 20 gal of hot water per minute.

Incinerators

For the destruction of soiled dressings and other refuse, one of the many types of steel or cast-iron garbage burners may be installed near the boiler room. Such incinerators are also constructed of brick or masonry.

With all types a supplementary fire must be maintained whenever the incinerator is in operation. They must be equipped with two separate grates or two sections of the same grate, so that refuse to be burned is not mixed with the fire maintained. When arranged for heating water, they are quite satisfactory for use when refuse is not being burned.

Incinerators for destruction of garbage in the basement are not recommended. The incinerators used in a number of United States public health hospitals, are constructed for one ton per day and is preferably used out of doors, but it can be made adaptable to nearly any condition.

When boilers are located in a separate building, the incinerator is frequently placed there and so arranged as to be charged from a platform outside the boiler house but fired from inside. Thus the care of it is placed in the proper hands, and the smoke pipe can be connected to the main chimney, eliminating, in this way, the principal objection to locating the incinerator near an inhabited building.

DISCUSSION

C. G. BINDER (WRITTEN): The society is to congratulate Mr. Russell for the splendid and painstaking paper he has presented. The information contained therein will be very helpful to the designing engineer who has occasion to lay out laundry, kitchen and hospital equipment.

There are several points upon which special emphasis should be placed, so that the equipment used will give maximum output and work at the highest

efficiency. As the equipment considered uses steam, it falls to the lot of the heating engineer to make provision for furnishing steam and to remove the condensation and air from the equipment.

I have had occasion to do some work in a laundry where a 5-roll flatwork ironer was installed, and on which one trap of sufficient capacity was used to drain it of condensation. The machine was loaded to its capacity, and the management was about to replace it with a 7-roll machine when a suggestion was made to change the method of draining the condensation. The recommendation was to install a thermostatic trap on each of the four chests and the last roll which was steam heated. This change being made permitted the more rapid and complete air removal which insured the maximum temperature within the equipment at all times, and increased the output of the machine in goods ironed by 30 per cent. This is merely one example of many that could be cited to show the advantages of thermostatic traps when properly applied to each coil or unit as was brought out in Mr. Russell's paper.

Reference is also made to the location of the supply and return mains. Frequently installations are made so that the return main is run at the ceiling. The author of the paper states that when the return main is at the ceiling it is necessary to use a mechanical trap. I must take exception to this statement. Thermostatic traps which handle automatically both the condensation and air are preferable to mechanical traps, irrespective of the location of the return main, *i. e.:* whether below or above the equipment, the only consideration being that the static head through which the condensation is lifted must not exceed the steam pressure. As in most cases, the steam pressure far exceeds the static head of the return above the equipment—this is of little concern.

It may be of interest to know that it is immaterial where the trap is located, *i. e.:* whether it be near the floor under the equipment, or at the ceiling where the return main is run. In either case the results will be entirely satisfactory.

J. D. CASSELL (WRITTEN): We are at this time, in Philadelphia, erecting a parental school in which nearly all of the subject matter is included, and I did think as I was perusing the paper on laundry equipment, that the author had omitted one of the most essential pieces of equipment for a laundry, which is a water-softener; but as I went further along, I found he included same, so that about passed up laundries.

Kitchen equipment, as referred to in the paper, is most complete, and being so much more extensive than anything we use, I failed to find anything omitted that would equip a kitchen completely.

With reference to the hospital, there I was all at sea and called to my assistance the Board's head physician, he in turn bringing with him two doctors competent in hospital work and equipment. They in turn went over the paper very minutely, and made these few suggestions:

First, to emphasize the thorough sterilizing of all dishes, both for dining-room, sick-rooms, and wards. The author has touched on all of these subjects, but they feel he did not make it weighty enough.

To insure against contamination of sterilized water by inleakage from the service water pipe, it was suggested that two stop valves be placed on this pipe adjacent to the sterilizer with a drain valve to an open cesspool, so that when the feed water is closed off, the drain valve could be open to insure against any inleakage.

They recommended that for a tuberculosis hospital, special incinerator be provided for sputum cups. Also, that swinging doors for hospital dining-rooms should be of the automatic type; either to open or close by electric contact or by weight of the person passing through, so that both hands can be used in carrying of wares, and the doors open upon the near approach of the parties passing through same.

Operating room doors should be provided with either the same device or an arm hook to allow opening without the sterilized hand touching knob.

A laundry chute should be provided for each ward to avoid carrying soiled bed and personal clothing out of the particular ward, these chutes to be lined with glazed brick or tile.

The hospital signal system should be by light rather than sound, these lights to be in plain view of head floor nurse. Further, that each bed should have one electric push button to summon nurse by the light system, so wired as not to be cut off by the nurse, only at the bed of the patient; and also registered on a board within the range of the head nurse.

An additional electric socket should be provided to attach electric pad, and one socket provided to attach radio for head phones. A three-wire electrocardiograph outlet should be installed in each ward from the cardiac room, so located that if the run to the furthermost bed were too great, any bed might be wheeled in place to the outlet.

Ample forced ventilation should be provided, including large warm air inlet ducts, to clear vapor in the main sterilizing room. In each ward there should be provided a small warmed wall closet for keeping laboratory specimens at bodily temperature until ready to be removed to the laboratories.

A separate inclosure should be provided in the maternity ward, constructed of glass to within three feet of the door for general observation purposes. This room to be heated to a temperature between 85 and 95 F with a heated closet for warming blankets, to be of ample size to be used not only for taking care of new born babies, baths, etc., but also those of premature birth.

While most of these latter subjects I have referred to, were not intended to be touched on by the author of the paper, our doctors thought them of enough importance to be included, or at least mentioned along with the paper.

I wish to congratulate Mr. Russell on what I consider a very worth-while contribution to this meeting, as his efforts are not only a paper but rather a compilation from which any engineer could most readily form a specification.

A. S. Kellogg (Written): It is a pleasure to review Mr. Russell's paper, constituting as it does, a distinctly valuable contribution to the files of the busy heating engineer who may at times be called upon to take into consideration such equipment as is here described; to clearly understand its proper functions and to properly locate and connect it for use.

Such remarks as are here made must not be construed as in any sense critical, but are intended to amplify the paper.

Under *Proportioning Equipment,* is there not a present tendency to get away from the small washer installations into sizes where a greater amount of work may be turned out at a relatively lower equipment cost and of greatly increased capacity? To be sure, one small machine is needed, but instead of a number of

small washers as were generally found in the flat work laundry, economic considerations are dictating a change to larger equipment.

As to *Washing Machines,* I hold that the wood washer has no place in the modern hospital or institution laundry, no matter how small the plant. Wood at its best is porous, and after a few years use seams and cracks offer opportunity for an unsanitary condition, and today aseptic rather than antiseptic conditions are recognized as of paramount importance. This is not generally afforded by the wood washer. The all-metal washer, preferably of monel metal, is coming more generally into use and rightfully so.

Is there not a printer's mistake in that paragraph relating to washer capacity? Surely the modern washer will average day in and day out a capacity at least double that stated, and the larger machines will quite easily take care of 5 lb of dry laundry per cubic foot per hour cycle.

In the past many mistakes have been made in installing too small flat work ironers, resulting in the slowing up of laundry capacity, with a greater amount of labor and expense for steam. Personally, I do not recommend ironers of less than 100 in. width having six rolls and a steam pressure of from 90 to 100 lb at the machine.

The modern *Dry Tumbler* is rapidly replacing the drying room of the truck and conveyor types in the strictly flat work laundry. Of course, where there is much starched work, collars, etc., to be handled, there the well-known dryroom has a rightful place.

The dry tumbler will leave blankets in a much better condition than the flat work ironer or dryroom, and it is also the writer's experience that the modern machine will not require in excess of 400 lb of steam per hour.

As to water, steam and power requirements, the paper well emphasizes the need of a generous supply of each. Except in those hospitals where it is insisted upon that the washers be brought to a boil for sterilizing reasons, the open end steam pipe, uncontrollable as it is, is to be avoided. It is much better for washing purposes that reliance be placed upon a generous supply of hot water. By so doing, marked economy in plant operation will result.

The following schedule of steam, water and power needs in hospitals will not fall outside good practice:

Washers	2 to 8 hp
Washers, cold water	100-gal per 100 lb dry laundry
Washers, hot water,	300-gal per 100 lb dry laundry
Extractors	2 to 10 hp
Tumblers	3 to 10 hp
Tumblers,	275 to 400 lb steam per hour
Flat Work Ironer	1 to 5 hp
Electric Irons	500 to 1000 watts

One also cannot be too generous in his demand factor.

It will not be amiss to mention the possibility of reclaiming a good portion of the heat from the waste from the washers by the installation of a proper heat exchanger. A marked economy in plant operation will result from so doing.

A new development in washer and extractor design should interest the very large hospital or institution.

It consists of what may be styled a *super-washer,* which by an appropriate power mechanism automatically dumps the load into the 3-part removable basket of the extractor. After the extraction process is completed the load is dumped as a whole onto the sorting table.

In addition to the washer and extractor, a small electric hoist and overhead track of simple design covers the needs of the installation.

Kitchen Equipment. I do not know of any equipment so flexible in meeting demands that may be made upon it as kitchen equipment.

The ventilation of some kitchens, however, may at times present a real problem. It is not at all unusual to require a change of air of once a minute, where cross ventilation of the room is impossible. It is desirable to remove all the air through the hood of the range or through several hoods if such there be.

The two precautions are, to see that the cooks cap is well anchored in place, and that free swinging doors are provided to the serving room and grill or dining room for the free ingress of the required air supply.

The fan power may run up to 50 hp or more in many cases.

Hospital Equipment. The section is excellently written. Here, however, the whims of the hospital staff of physicians or the superintendent may be expected to assume full tempermental sway, sometimes with unsatisfactory results.

May I suggest one other piece of equipment to round out the list? That is, the oil instrument sterilizer in which oil of Alboline grade is used. This should be heated to 300 F, which temperature is considered necessary for destroying spores. That is just a new one to be-devil humanity.

Needless to say that the oil bath should be provided with an automatically controlled electric unit. Those I have so far seen in use require 1000 watts, and as there is practically no evaporation from the surface of the oil, the danger of burning out the element is remote.

E. B. ROYER (WRITTEN): This paper contains much valuable information often desired but very difficult to obtain and is greatly appreciated by consulting engineers and others who are called on to specify the mechanical equipment of hospitals and institutions.

As the data contained will be used so widely and will no doubt be reprinted in THE GUIDE next year, any general statement which might lead a fellow engineer into trouble should be qualified. I am, therefore, offering the following:

In the section referring to steam pressures for kitchen equipment, the paper states that "not less than 30 lb gage and not over 50 lb are the usual pressure limits for steam cooking" and that "all jacketed pots have safety valves usually set at 50 lb." This setting of the safety valve at 50 lb was true for the tin-lined copper-jacketed kettle formerly used almost exclusively but the aluminum steam-jacketed kettles now more commonly used, partly on account of their lower cost, are designed for but 40 lb pressure. Quoting from the catalogue of one of the leading manufacturers, their aluminum steam-jacketed kettles are "designed to carry not over 40 lb but can be used on much less. In case more than 40 lb is carried on supply lines, a reducing valve should be used." Similar data appear in the catalogue of other manufacturers.

Properly designed kitchen equipment (with a very few exceptions such as possibly oyster stewers) can be operated satisfactorily on pressures as low as

12-15 lb obtained from the standard, highly efficient, cast-iron sectional gas-fired boiler. This has been demonstrated by the use of these pressures repeatedly in several chains of well-known restaurants with good results.

It, therefore, appears that in future reprints of this paper and later editions of THE GUIDE, 40 lb should be specified as the safe, usual upper limit for steam pressures for kitchen equipment in general unless regular aluminum steam jacketed kettles are avoided.

No. 821

INVESTIGATION OF HEATING ROOMS WITH DIRECT STEAM RADIATORS EQUIPPED WITH ENCLOSURES AND SHIELDS

By A. C. Willard[1] (*Member*), A. P. Kratz[2] (*Member*), M. K. Fahnestock[3] (*Non-Member*) and S. Konzo[4] (*Non-Member*)

Urbana, Ill.

Introduction

THE data presented in this paper were obtained in connection with an investigation which is being conducted by the Engineering Experiment Station of the University of Illinois, of which M. S. Ketchum, Dean of the College of Engineering, is Director, in cooperation with the *National Boiler and Radiator Manufacturers' Association* and the *Illinois Master Plumbers' Association,* under the supervision of A. C. Willard, professor of heating and ventilation and head of the department of Mechanical Engineering. This paper constitutes a partial summary of results which are to be presented in complete form as Engineering Experiment Station Bulletin No. 192 entitled Investigation of Heating Rooms with Direct Steam Radiators Equipped with Enclosures and Shields.

The results presented in this bulletin are based upon the work done since the publication of Bulletin No. 169 entitled Effect of Enclosures on Direct Steam Radiator Performance, in which were reported the results of the first year's work under this agreement.

Object of Investigation. The immediate object of the test reported in part in this paper was to determine the effect of various types of present-day commercial radiator enclosures, shields and covers on the heating effect produced and the steam condensed by a direct cast-iron radiator placed in an actual room subjected to zero weather conditions.

Scope of Investigation. The effect of an enclosure, shield or cover upon the

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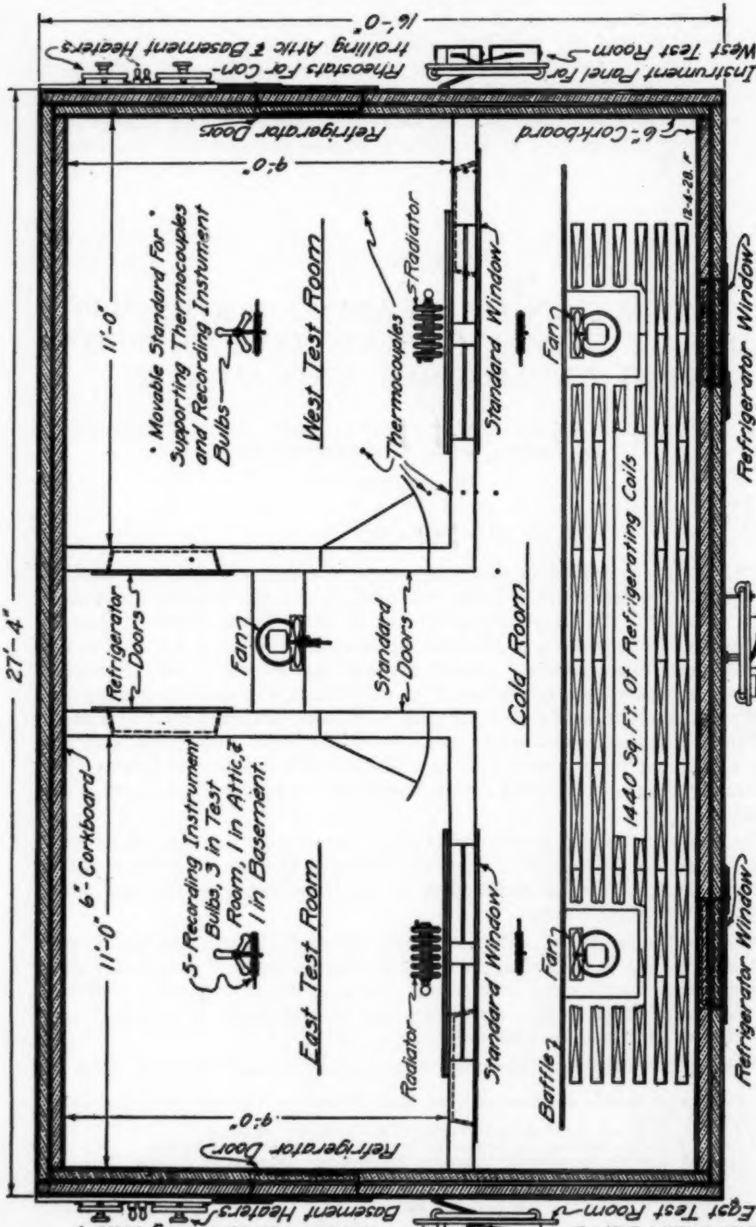
Presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Chicago, Ill., January, 1929.



Note:-Thermocouples shown as •, Recording Thermometer Bulbs shown as —

FIG. 1. PLAN SECTION OF LOW TEMPERATURE TESTING PLANT

heating effect produced in a room and the steam condensing capacity of a radiator depends upon many factors. The tests made in connection with the present investigation were planned to determine the influence of all of the factors which enter into this problem in the case of various commercial radiator enclosures and shields. In conjunction with the work on enclosures and shields, tests were run on an unenclosed radiator, a variety of cloth covers and a special shielded radiator.

In order to provide for all the factors affecting the performance of bare, enclosed and shielded radiators, actual rooms with typical outside walls, windows and doors, and located in a specially constructed low temperature testing plant for maintaining constant outside temperatures of zero or less were used. In such a plant, it was possible to place the radiator in the actual environment existing in practice, and investigate not only the heat emission of the radiator itself, but also the heating effect produced in the room as well. The intelligent design and selection of radiators and enclosures depends fully as much on the effect produced in the room as on the conventional heat emission factor so generally taken as the sole criterion of excellence in the past.

The investigation which is reported in part in this paper is an elaborate extension of the previous investigation in this field, the results of which were published in Bulletin No. 169[5]. The latter investigation was confined strictly to the heat emission or steam condensing capacity of bare and enclosed radiators under the usual laboratory conditions. The correlation of the results of that investigation with this investigation is entirely satisfactory.

DESCRIPTION OF APPARATUS

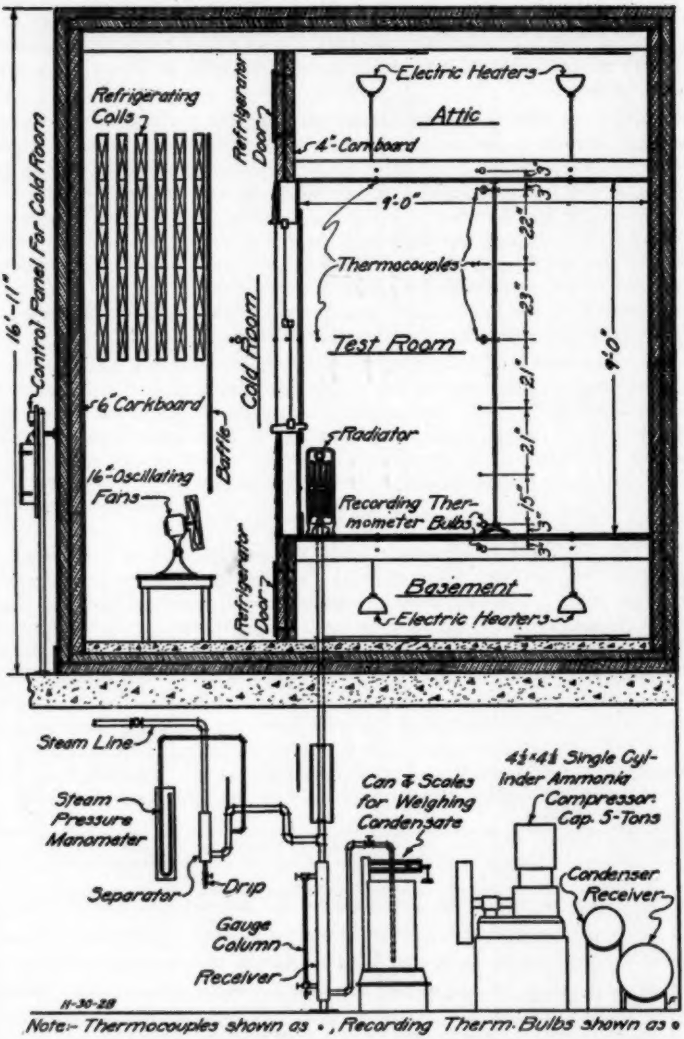
Description of Low Temperature Testing Plant. The plant is designed specifically for the purpose of accurately studying direct steam and hot water heating problems, including those phases of building construction and insulation which are of special interest to the heating contractor and engineer, as well as the building owner and manufacturer of heating equipment, under conditions approaching, as nearly as possible, those found in actual practice.

The general arrangement of the plant is shown in Figs. 1 and 2. The main portion of the plant, consisting of the cold room, the two test rooms with their respective attics and basements, and the refrigerating coils, is located on the upper floor of the Mechanical Engineering Laboratory. The auxiliary equipment including the refrigerating apparatus, with the exception of the coils, the

[5] The two cooperating associations have been represented since the publication of the first bulletin by an advisory committee, the membership of which is as follows:

C. D. Brownell, Chairman, representing the *Illinois Master Plumbers' Association*, Champaign, Ill.
C. A. Bolton, representing the *Illinois Master Plumbers' Association*, Chicago Heights, Ill.
C. K. Foster, representing the *National Boiler and Radiator Manufacturers' Association*, Chicago, Ill.
H. R. Linn, representing the *National Boiler and Radiator Manufacturers' Association*, Chicago, Ill.
R. F. Prox, representing the *National Boiler and Radiator Manufacturers' Association*, Terre Haute, Ind.
F. W. Herendeen, representing the *National Boiler and Radiator Manufacturers' Association*, Geneva, N. Y.
O. J. Prentice, representing the *Steam Specialties Manufacturers*, Chicago, Ill.
H. S. Ashenhurst, representing the *Insulation Manufacturers*, Chicago, Ill.
W. H. O'Brien, representing the *Lumber Industries*, Chicago, Ill.
Seward Best, representing the *Heating Contractors*, Quincy, Ill.
J. M. Robb, representing the *Heating Contractors*, Moline, Ill.
H. F. Burch, representing the *Heating Contractors*, Rock Island, Ill.
J. F. Powers, representing the *Heating Contractors*, Springfield, Ill.

thermocouple switchboard and the steam control and weighing apparatus, is located on the lower floor of the laboratory.



Note:- Thermocouples shown as •, Recording Therm. Bulbs shown as •

FIG. 2. ELEVATION SECTION OF LOW TEMPERATURE TESTING PLANT

The Test Rooms. Figs. 1 and 2 show the arrangement of the two test rooms which are identical in construction, each one having two walls exposed to the

air in the cold space in which the refrigerating coils are located. The cross-hatched walls in Figs. 1 and 2 are composed of corkboard. The two exposed walls of both test rooms, indicated by light lines, can be removed and replaced with any desired wall construction without disturbing the floors or ceilings. As shown in these figures, the insulated walls of the cold room form the two remaining walls of each of the test rooms. Both test rooms are identical in every detail, being 9 ft x 11 ft with 9-ft ceiling heights. The exposed walls at the present time (Fig. 2a) are standard frame construction, consisting of ⅝-in. redwood siding, building paper, ¾-in. tongue and groove yellow pine sheathing, 2 x 4 yellow pine studding, and ⅜-in. wood lath with ½-in. gypsum plaster. The ceilings are made of ⅜-in. wood lath and ⅜-in. gypsum plaster,

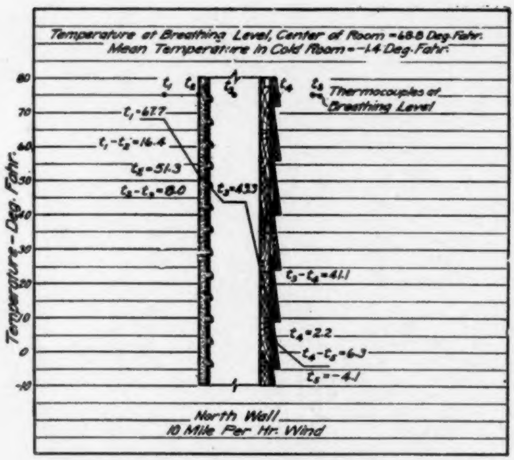


FIG. 2a. TYPICAL TEMPERATURE GRADIENT THROUGH NORTH WALL OF WEST TEST ROOM, TEST R-9

with no flooring in the attics. The floors are of standard 2¼-in. x 13/16-in. standard yellow pine flooring over building paper placed on ¾-in. thick tongue and groove sub-floors.

Fig. 4 shows the inside of both test rooms, with radiators in front of the windows, which are placed in exposed walls as shown in Figs. 1 and 2. Each test room has one double window 4 ft 6 in. x 5 ft overall. The window stools are 34 in. high, making it possible to test radiators with a height up to 32 in. The windows are fitted with shades and curtains as shown in Figs. 4 and 7. Figs. 1 and 4 show the locations of standard 1¾-in. thick yellow pine doors, 3 ft x 7 ft, with glass upper panels. These doors lead directly from the test rooms into the cold space.

Fig. 2 shows the attics and basements located above and below each of the test rooms. These attics and basements are for the purpose of exposing the ceilings and floors of the test rooms to air of any desired temperature. The

attics were formed by building the ceilings of the test rooms about 3 ft 10 in. below the ceiling of the cold room, and the basements were made by building the floors of the test rooms about 2 ft 8 in. above the floor of the cold room. The exposed walls of the attics and basements are composed of 2 layers of 2-in. corkboard, and are equipped with refrigerator doors as shown in Fig. 5. Each wall contains one door, located as shown in Fig. 1, which opens directly into the

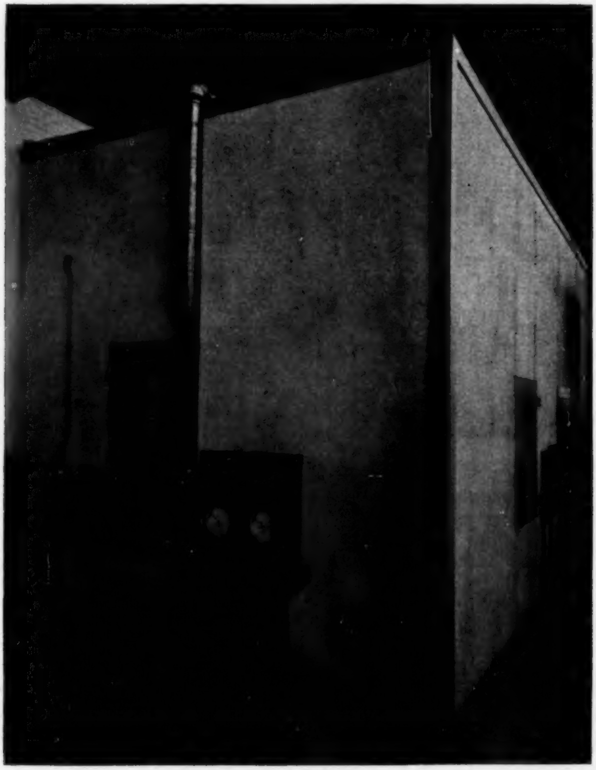


FIG. 3. OUTSIDE OF COLD ROOM, SHOWING RHEOSTATS, RECORDING INSTRUMENTS AND AUTOMATIC EXPANSION VALVE

cold area, and which may be adjusted to obtain any desired amount of opening. Each attic and basement is equipped with electric heaters, as shown in Fig. 2, consisting of shaded electric light bulbs, so placed that the heat is evenly distributed over the total floor and ceiling surfaces and shielded in order to prevent the floors and ceilings from receiving heat by direct radiation. The voltage to each group of heaters in each attic and basement is separately controlled by

rheostats conveniently located outside of the cold room, as shown in Figs. 1 and 3. These heaters, so arranged and controlled, when used in conjunction with the refrigerator doors give a very flexible and sensitive method of controlling the temperatures in the attics and basements.

The Cold Room. In order that two walls of the test rooms might be exposed to conditions corresponding to those prevailing during the heating season, it was necessary to enclose the test rooms in a cool space in which the temperature and wind movement could be controlled. This makes it possible to run tests at any time of the year under exactly similar conditions. Fig. 3 shows the outside of the cold room, which is 16 ft x 27 ft 4 in. x 16 ft 11 in. in height. Figs. 1 and

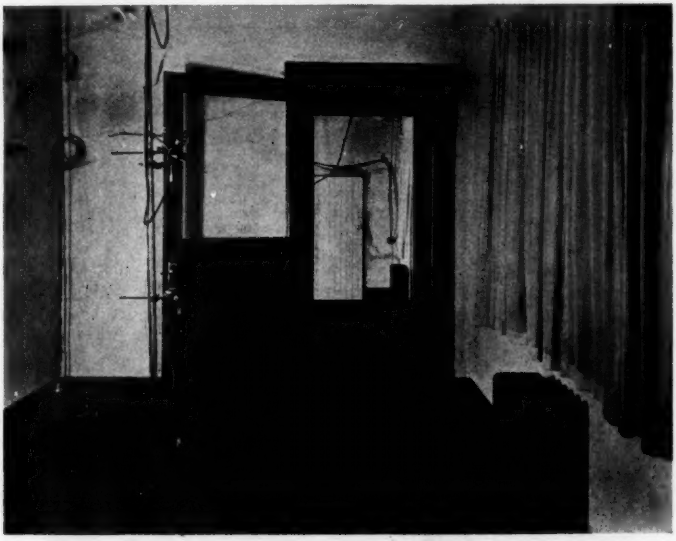


FIG. 4. INSIDE VIEW OF BOTH TEST ROOMS

2 show the construction details. The walls consist of 2 layers of 3-in corkboard, with ½ in. of cement mortar between them, and ½ in. of cement plaster on the inner and outer surfaces. The ceiling is made of 2 layers of 3-in. corkboard laid in hot asphalt on a ¾-in. wood deck which is supported independently from the walls of the room. The floor consists of 4 in. of concrete laid on 6 in. of corkboard, which in turn is laid on the 10-in. concrete floor of the laboratory. Entrance to the cold room is through either test room as shown in Fig. 1. The doors, leading from the laboratory into the test rooms, are of the heavy refrigerator type as shown in Fig. 3. Two refrigerator windows, having four separate sheets of glass and three air spaces in each, are located in the north wall as shown in Figs. 1 and 3.

Wind movement in the refrigerated space is obtained by means of three 16-in., specially built, oscillating fans located as shown in Figs. 1 and 2.

Refrigerating Apparatus. The plant is designed for an operating temperature of zero deg F in the cold room when the temperature at the breathing line in the test rooms is 70 F. This temperature in the cold space is maintained by

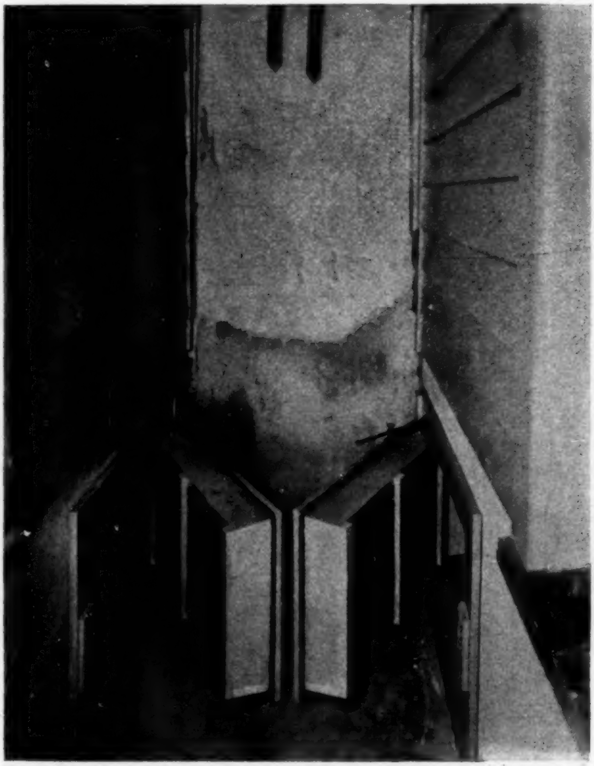


FIG. 5. REFRIGERATOR DOORS OPENING INTO BASEMENT UNDER BOTH TEST ROOMS

means of a 5-ton direct expansion ammonia refrigerating unit of the compressor type, located in the basement of the laboratory, as shown in Fig. 2. The compressor, which is motor driven and automatically controlled by a thermostat placed in the cold room, is shown in Fig. 6 together with the ammonia condenser and receiver. The ammonia is expanded through an automatic expansion valve directly into the coils shown in Figs. 1 and 2. These coils are special cast-iron refrigerating sections having a total area of 1440 sq ft. A baffle, shown

in Figs. 1 and 2, was erected between the coils and the test rooms for the purpose of increasing the air circulation over the coils and to shield the walls and windows of the test rooms from *direct radiation.*

Recording Instruments and Thermocouples. The type of test conducted in this plant necessitates the accurate duplication and maintenance of conditions over comparatively long periods of time.

The selection and installation of apparatus, therefore, had to be based on considerations relative to the adaptability for controlling conditions as well as to accuracy of observations.

Figs. 1, 2 and 3 show the location of the recording instruments which were

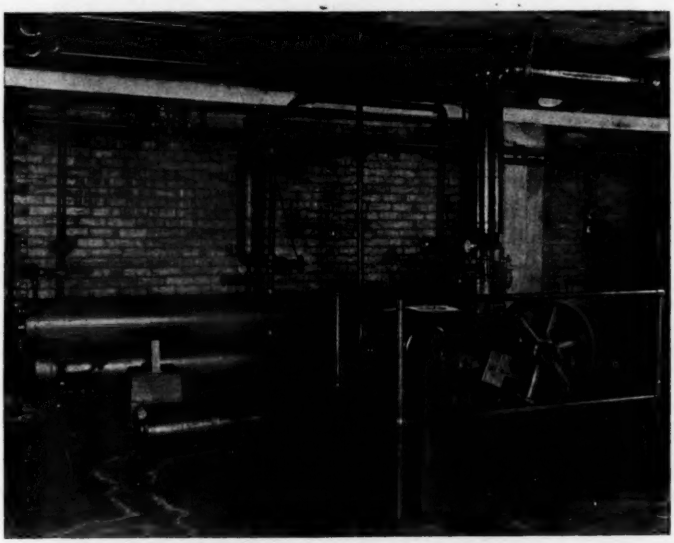


FIG. 6. REFRIGERATING APPARATUS, INCLUDING THE AMMONIA COMPRESSOR, THE CONDENSER AND RECEIVER

installed for the purpose of control. Each test room, and the cold room, have separate instruments and instrument panels. Fig. 7 shows the inside of one of the test rooms with the standard in the center of the room supporting the recording instrument bulbs and thermocouples. Three of these bulbs are located in each of the test rooms; one 3 in. above the floor, one at the breathing line and one 3 in. below the ceiling. In the corner of the test room, Fig. 7, may be seen leads going to the recording instrument bulbs in the attic and basement. One bulb is located in the center of each attic and basement, 3 in. above the ceiling and 3 in. below the floor. Fig. 2 shows the location of the three recording instrument bulbs in the cold room, each of which is placed at a height corresponding to the breathing level of the test rooms.

Fig. 7 also shows the apparatus used for generating fumes and making studies of the air circulation in the test rooms. It consists of a portable standard, supporting three pairs of concentric glass dishes which contain ammonium hydroxide and hydrochloric acid. The vapors from these two chemicals, when allowed to mix, form dense white fumes of ammonium chloride. The clamps and platforms holding the glass dishes may be adjusted on the standard in order to place the dishes at any desired level.

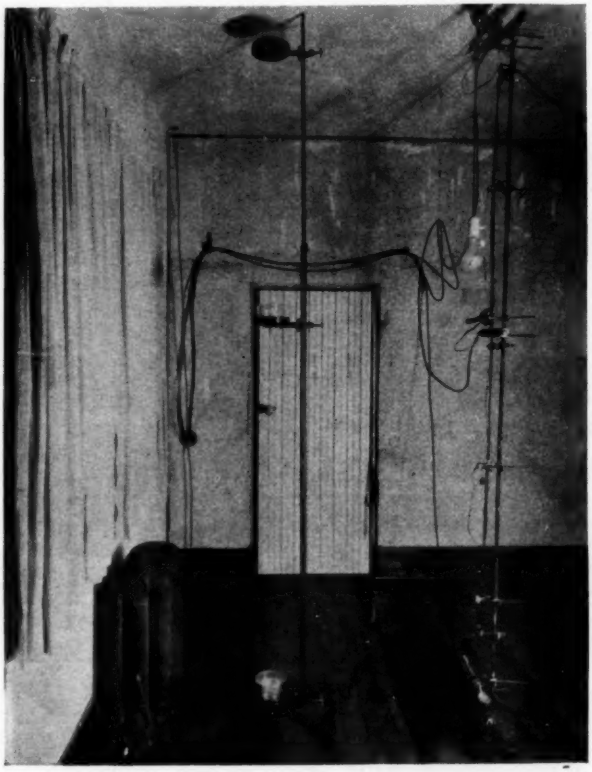


FIG. 7. INSIDE OF EAST TEST ROOM SHOWING SMOKE STUDY APPARATUS AND STANDARD SUPPORTING RECORDING INSTRUMENT BULBS

The plant is equipped with a complete thermocouple system for the purpose of observing both air and surface temperatures. The use of thermocouples makes it possible to obtain the necessary temperature data without entering the rooms or otherwise disturbing the test conditions. They also make it possible to observe surface temperatures and certain air temperatures which could not

be obtained accurately by means of thermometers. The thermocouples are made of No. 22 B. & S., double cotton covered copper and constantan wire. The leads from all the couples are formed into cables and connected to the switchboard shown in Fig. 8, which is located on the lower floor of the laboratory directly beneath the cold room. Fig. 7 shows one of the standards supporting six thermocouples, used in determining air temperatures at six different elevations in the center of the test rooms. The height at which each one of these thermocouples is located is shown in Fig. 2.

Figs. 1 and 2 show the location of thermocouples by means of which the temperature gradients through the walls, floor and ceiling of the west test room were determined. These figures also show the thermocouples used in determining the temperature of the inside and outside surfaces of the wall back of the radiator, and the thermocouples located at each recording instrument bulb for the purpose of checking and adjusting these instruments. All thermocouples for observing surface temperatures had the junctions, and approximately 4 in. of the leads on both sides of the junctions, embedded in the surfaces. The wires were placed in a deep scratch in the surface, and were sealed into the surface itself by means of plaster of Paris in the case of plaster surfaces, and shellac in the case of wood surfaces. The wires were then filed flush with the surface and thus became an integral part of it.

The Weighing System. The condensate weighing system, and the method of regulating the pressure of the steam in the radiators or heating units in the test rooms, is similar to that used in the tests published in Engineering Experiment Station Bulletin 169. As shown in Fig. 2, the piping, separator, receiver, weighing tank and scales are placed in the basement of the laboratory directly beneath the test rooms. Each test room is piped separately and is fully equipped to be operated independently of the other one. Separators are used to remove all entrained moisture from the steam, and mercury manometers are used to indicate the steam pressure. The temperature of the steam is observed by means of thermocouples just before it enters the radiator. Glass sections, 1⁵⁄₁₆-in. inside diameter, are installed in the 1¼-in. vertical risers to the lower tappings of the radiators. The condensate leaves the radiators through these same connections, and is collected in receivers having gage columns. The weighing tanks are connected through water seals to the receivers, and the minimum subdivisions of the scales used for weighing the condensate is 0.01 lb. The separators, receivers and piping are all heavily lagged, and the glass sections in the vertical risers are enclosed in triangular glass observation boxes for protection and prevention of heat loss. See chapter on Test Methods for calibration and corrections for all piping losses. Each radiator is equipped with a ⅛-in. pipe leading from the tapping for the lower air vent on the last section to the lower floor of the laboratory, where the amount of venting is controlled by means of hand-operated gate valves.

Description of Unenclosed Radiators. The unenclosed radiators used for the tests included in this paper were 6-section, 26-in., 5-tube, cast iron radiators having rated areas of 21 sq ft. The surfaces were brushed and painted (not dipped) with two coats of flat black paint. Fig. 9 shows one of these radiators located under the curtained window with a space of 2½ in. between the back of the radiator and the plaster surface of the exposed walls.

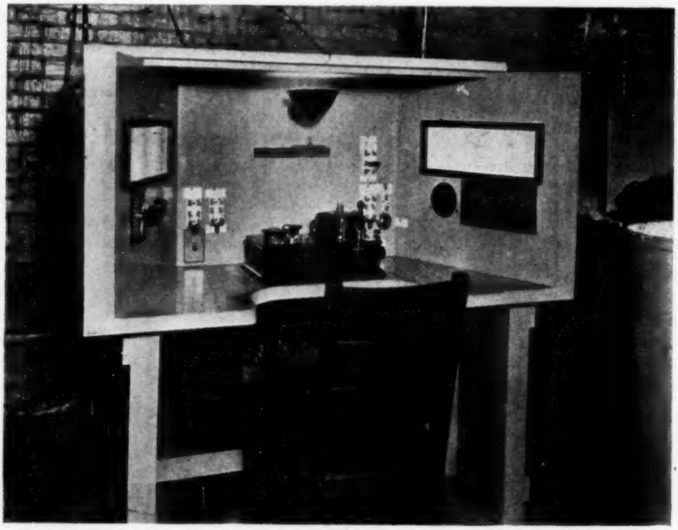


FIG. 8. THE THERMOCOUPLE SWITCHBOARD WITH A PRESENT CAPACITY FOR 122 THERMOCOUPLES

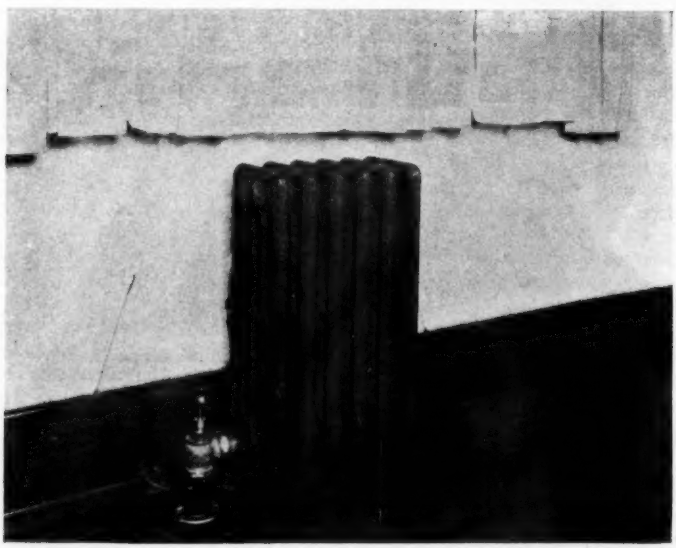


FIG. 9. UNENCLOSED RADIATOR

Descriptions of Enclosures. Figs. 10 to 13 show the enclosures, covers and shields tested. In all tables, and in the text, each piece of apparatus tested, whether it is an ordinary enclosure or a cloth shield, is designated as an enclosure. In order to differentiate between them and to simplify the presentation and discussion of results, each enclosure is numbered as shown in the figures.

Enclosure No. 1, shown in Fig. 12, is a common commercial type of metal shield.

Enclosures Nos. 3, 10 and 11, shown in Figs. 10 and 11, are commercial metal enclosures of different types. It should be noted, that the clearance between the top of the radiator and the bottom of the humidifying pan is practically the same in each case. The inside widths and lengths of these various enclosures vary a small amount.

Enclosure No. 4, shown in Fig. 13, is a crash cloth cover fitted over the top of the radiator.

Discussion of Test Methods and Results

Limiting Conditions for Tests. All tests were run under conditions approximating those found in typical rooms in residences, with two walls exposed to an outdoor temperature slightly below zero and with some wind movement over one wall. The motor-driven ammonia compressor was thermostatically controlled, and the temperature in the refrigerated space at a level corresponding to the breathing level in the rooms was automatically maintained at approximately —1.5 F.

The temperatures in the air spaces above the ceilings and below the floors of the test rooms were regulated by means of electric heaters and rheostats under manual control. These heaters were shielded in order to minimize the effect of direct radiation on the surfaces of the floors and ceilings. The temperature of the air 3 in. above the ceilings was maintained at approximately 62 F, while that 3 in. below the floors was maintained about 2 F higher than the temperature of the air 3 in. above the floors. In certain cases, temperatures in actual attic spaces may be somewhat lower and temperatures immediately under the floors may be somewhat higher, but the conditions selected correspond very closely with those found with a well-constructed roof and unfloored attic, and with a well-insulated heating plant, where the basement temperature is approximately 60 F.

The amount of standard 5-tube radiation required to maintain a temperature of 70 F at the breathing level, or 5 ft above the floor under the conditions outlined, was determined by trial from preliminary runs. This amount proved to be 21 sq ft for the bare radiator, and remained unchanged for all of the tests. The various enclosures, shields and covers were made to fit this radiator, and the performance of the bare radiator was used as the basis for comparison. When enclosures, shields or covers were installed, no other changes were made in the plant or in the temperature conditions external to the rooms. In case an enclosure was equipped with a humidifying pan, the pan was retained in its proper location but no water was used. The temperatures at the various levels within the rooms were allowed to attain equilibrium conditions inherent with the type of apparatus being tested, and comparisons of the performance were all made on the basis of steam condensation in conjunction with the temperature

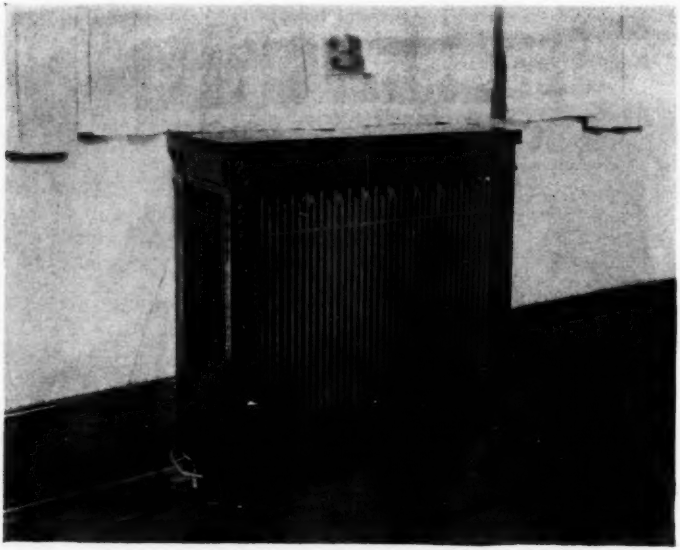


FIG. 10. ENCLOSURE NO. 3

FIG. 11. ENCLOSURE NO. 10

conditions produced within the rooms as determined at six different levels. See Figs. 1 and 2.

Temperature Measurements. The temperature of the air in the rooms was observed by means of thermocouples placed at six different levels on the central vertical axis of the rooms. These couples were of No. 22 B. & S. gage wire and were unshielded. Preliminary work with shielded and unshielded couples proved that no correction for radiation was necessary for the couples above the breathing level, and that the maximum correction for any couple below the breathing level was less than 0.5 F. Hence the readings of the unshielded couples are considered as indicative of the actual air temperatures. Temperatures of the outside and inside surfaces of the walls were obtained by means of 6 thermocouples (Figs. 1 and 2) embedded in the surfaces. Similar temperatures of the upper and lower surfaces of the floor and ceiling were obtained at four different points (Figs. 1 and 2) on each surface.

Operation of Plant. Both rooms were operated simultaneously, and in every case check runs were made with the same enclosure, first in one room and then in the other. These check runs proved that the performance, both of the rooms themselves and of the apparatus in the rooms, was practically identical. For this reason, it has not been considered necessary to report the results of the duplicate tests. No attempt was made to have the doors and windows tighter than what could be considered fair average construction, and the amount of infiltration of cold air into the rooms appeared to be normal for the wind movement and temperatures existing.

In every case, the plant was operated with the rooms under heat and with the fans in the cold room running for a preliminary period of sufficient length to allow all conditions to attain a state of equilibrium. This state was indicated when the temperatures of the walls, floors and ceilings, as determined by the readings of the surface thermocouples had become constant, and the temperature of the air in the refrigerated space, and in the spaces below the floors and above the ceilings had remained constant for several hours. No air was allowed to accumulate in the radiators during the preliminary period or during a test. When equilibrium had been attained, the condensation from the radiators was weighed at 10-min intervals, and no test was accepted that showed a variation of more than 2½ per cent in the successive increments of weight. A thermocouple on the surface of the radiator, at the point where air accumulation would first occur, gave an immediate indication if there was any tendency for air to accumulate during a test. The tests were of sufficient length to prove that all conditions had remained constant, and were discontinued at the first indication that any air had accumulated in the radiator. The condensation was corrected for water condensed in risers and piping. This correction was determined from preliminary tests, and, since all piping was heavily lagged with hair felt, was extremely small.

The results of all tests here reported are given by means of the curves in Figs. 14 to 17. A discussion of these curves may be found under the corresponding section headings.

TESTS OF BARE RADIATORS

Results of Performance Tests. Several tests were run with two identical

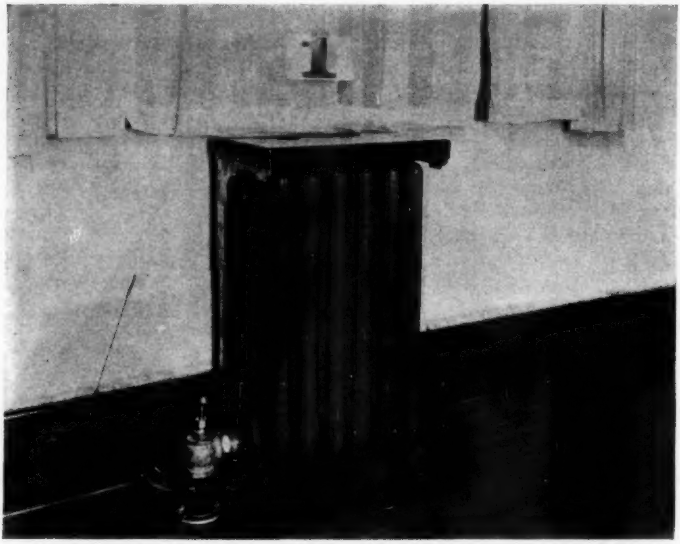


FIG. 12. ENCLOSURE NO. 1

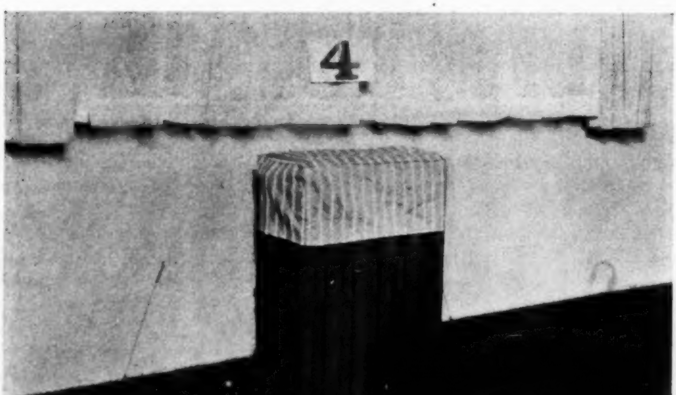


FIG. 13. ENCLOSURE NO. 4

bare radiators (Fig. 9) operated simultaneously in the two rooms. Other tests were run with a bare radiator in one room and an enclosed radiator in the other room. In all cases, the net steam condensed by the bare radiator per hour and the temperature gradients in the rooms in which the bare radiator was used were practically identical, provided that the control conditions were the same. Accordingly, a single curve was selected to represent the performance of the bare radiator in each room, and this curve has been reproduced in each set of curves in Figs. 14-17 corresponding to the room in which the tests were run in order to serve as a basis for comparison for the performance of the enclosed radiators. From these curves it may be noted that when an air temperature of 69.4 F was maintained at the breathing level in the West test room, and a steam temperature of 216.5 F was maintained in the bare raditor, the net weight of steam condensed per hour was 5.44 lb. Under these conditions, the temperature of the air 3 in. above the floor was 54.3 F and that 3 in. below the ceiling was 76.1 F, or a difference of 21.8 F. A fairly uniform temperature gradient in the air from floor to ceiling was obtained. With 69.4 F at the breathing level the whole zone below the breathing level, which may be regarded as the living zone, was too cool for satisfactory comfort. The high temperature at the ceiling resulted in an excessive heat loss through the ceiling itself.

Radiator Rating. The catalog rating for the type of bare radiator used, based on the Engineering Standard[6] of 240 Btu emission per square foot per hour with steam at 215 F in the radiator and air at 70 F surrounding it, was 21 sq ft. The actual superficial area of the radiator by measurement was 19.3 sq ft. The total heat emission under the test conditions, based on steam at 216.5 F in the radiator and a temperature of 69.4 F at the breathing level, was 5.44 x 969.1 = 5270 Btu per hour. Hence, the total heat emission under standard rating conditions would be $5270 \dfrac{(215-70)^{1.3}}{(216.5-69.4)^{1.3}} = 5180$ Btu per hour, and the rating as determined from these tests should be $\dfrac{5180}{240} = 21.6$ sq ft based on the Engineering Standard as compared to 21.0 sq ft given as the catalog rating. Therefore, the catalog rating is approximately correct if the radiator is used in a room with an air temperature of 70 F at the breathing level, and a steam temperature of 215 F.

Let K = the coefficient of heat transmission in Btu per sq ft per deg fahr difference in temperature between steam and air per hour. Then

$$K = \frac{5180}{19.3(215-70)} = 1.85$$ for the standard conditions based on measured surface.

TESTS WITH ENCLOSURES

Introduction. The results of tests with two characteristic types of commercial enclosures are shown in Figs. 14 and 15. The corresponding types are shown in Figs. 10 and 11.

In comparing the performance of the enclosed or shielded radiators with that

[6] JOURNAL OF AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Vol. 33, No. 3, March, 1927.

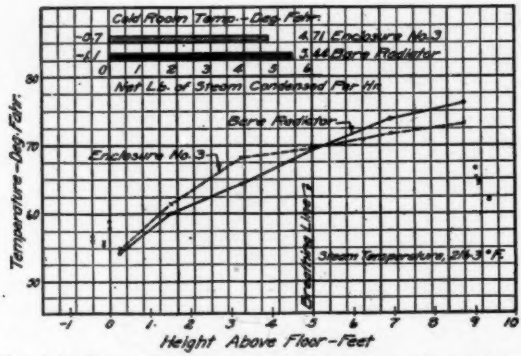


FIG. 14. ROOM TEMPERATURE GRADIENT AND STEAM CONDENSING RATE FOR RADIATOR WITH ENCLOSURE NO. 3. TYPE SHOWN IN FIG. 10

of the bare radiator, several factors in conjunction must be taken into consideration. These are: (1) the effect on the temperature at the breathing level, (2) the effect on the temperature at the ceiling, or the difference in temperature between the floor and ceiling, (3) the effect on the mean temperature in the living zone, or the zone below the breathing level, (4) the relative steam condensation per hour. In general, an enclosure may be regarded as better than the bare radiator if (1) a breathing level temperature within 1 F above or below 69.4 F was maintained, (2) if the temperature at the ceiling was lowered, (3) if the floor temperature was maintained equal or raised, (4) if the mean temperature in the living zone was raised and (5) if the steam condensation was the same as, or less than, that of the bare radiator.

RESULTS OF TESTS

Fig. 14 indicates that enclosure No. 3 shown in Fig. 10, as compared to the

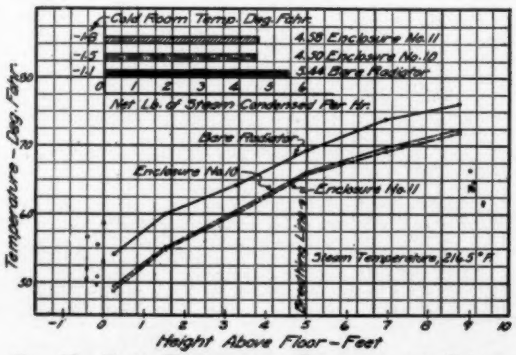


FIG. 15. ROOM TEMPERATURE GRADIENTS AND STEAM CONDENSING RATES FOR RADIATOR WITH ENCLOSURES NOS. 10 AND 11. TYPES SHOWN IN FIG. 11

bare radiator, maintained the same breathing level temperature, lower ceiling temperature, higher floor temperature and materially higher temperature in the living zone. The steam condensation was also materially less than that for the bare radiator. This enclosure was unquestionably superior to the bare radiator both from the standpoint of air temperature conditions in the room and of steam economy.

Fig. 15 shows the results obtained with enclosures Nos. 10 and 11. These enclosures were both of the same type, as shown in Fig. 11, except that No. 10 fitted the radiator snugly while No. 11 had relatively large side clearance. The performance of this type of enclosure was far from satisfactory. The steam condensation was very much reduced over that of the bare radiator. This re-

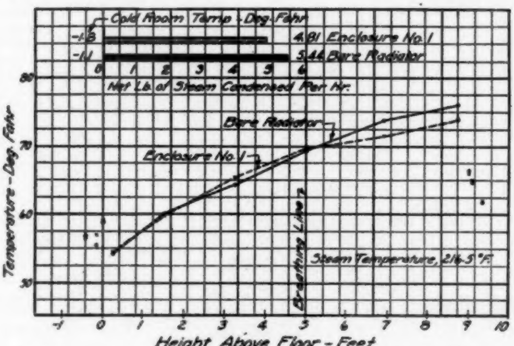


Fig. 16. Room Temperature Gradient and Steam Condensing Rate for Radiator with Enclosure No. 1. Type Shown in Fig. 12

duction can in no sense be regarded as an economy, however, because the enclosed radiator failed to heat the room. All of the air temperatures in the room were from 4 to 5 F lower than those obtained with the bare radiator. This condition is particularly objectionable near the floor and in the living zone as shown by the air temperatures in this zone. The results with the snugly fitting enclosure were slightly better than those for the one with large side clearance. This seems to indicate that on the whole large clearances are undesirable.

Tests with Metal and Cloth Shields

Metal Shield. The metal shield tested has been designated as enclosure No. 1, and the type is shown in Fig. 12. The results of the tests on this shield are shown in Fig. 16, from which it is evident that the use of the shield resulted in a lower temperature at the ceiling than that obtained with the bare radiator. The temperature at the floor was the same as, and the mean temperature below the breathing level was slightly higher than the corresponding temperatures for the bare radiator. The net steam condensation was less than that for the bare radiator. This shield, therefore, was more advantageous than the bare radiator in that it produced more satisfactory air temperature conditions accompanied by greater steam economy.

Cloth Covers. Tests were run with one type of cloth cover completely enclosing the upper part of the radiator. This cover is shown as enclosure No. 4 in Fig. 13. The results are indicated in Fig. 17. This cover reduced the steam condensation materially, and produced entirely unsatisfactory air temperature conditions in the room. The cover reduced the breathing level temperature to

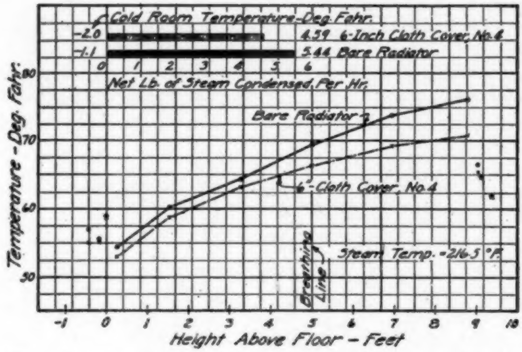


FIG. 17. ROOM TEMPERATURE GRADIENT AND STEAM CONDENSING RATE FOR RADIATOR WITH ENCLOSURE NO. 4. TYPE SHOWN IN FIG. 13

66.4 F. Corresponding reductions in the mean temperatures below the breathing level and in the temperatures at the floor were also observed. This cover had nothing to recommend it, and its use may be expected to result in underheated rooms, unless a correspondingly greater amount of radiation is installed.

COMPARISON OF THE PERFORMANCE OF THE VARIOUS TYPES OF ENCLOSURES

Performance of Enclosures. A comparison of the two types of commercial enclosures tested may be made from Table 1. From this table it is evident that the most satisfactory combination of all of the factors involved was obtained with enclosure No. 3. Since the degree of comfort produced is probably the final criterion for judging the performance of any given heating unit, the mean temperature in the zone from the floor to the breathing level is the most important factor involved. The highest mean temperature below the breathing level was obtained with enclosure No. 3.

On comparing the two enclosures in respect to structural differences, it may be noted that the one giving the more satisfactory results was the one having

TABLE 1. COMPARISON OF PERFORMANCE FACTORS FOR ENCLOSURES

Enclosure No.	Type Fig. No.	Performance Fig. No.	Temp. at Breath. Level Deg Fahr	Diff. in Temp. Ceiling-Floor Deg Fahr	Mean Temp. below Breath. Level Deg Fahr	Net Steam Condensed Lb/Hr
Bare	9	14-17	69.4	21.8	62.0	5.44
3	10	14	69.5	18.2	63.6	4.71
10	11	15	66.2	23.0	57.8	4.50

the greatest free area of openings, thus offering the least restriction to the flow of air over the radiator. This may be made clear by comparing enclosures Nos. 3 and 10 as shown in Figs. 10 and 11 in the order designated. This comparison immediately makes it evident that the reason for the unsatisfactory

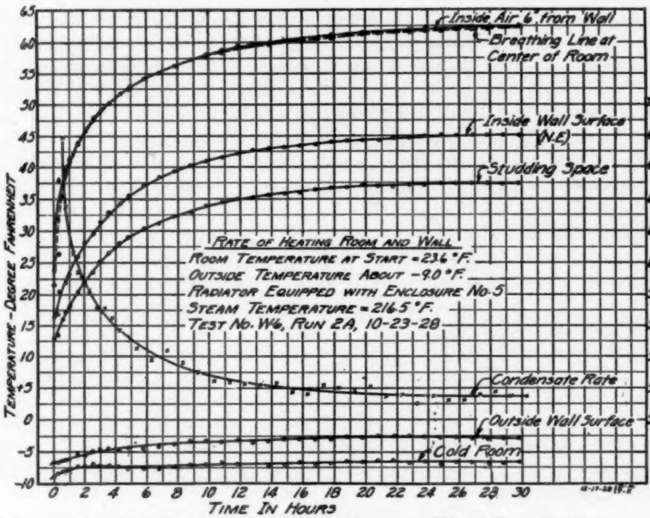


FIG. 18. TYPICAL WARMING CURVE FOR WEST TEST ROOM

performance of enclosure No. 10 is that the total area of the grilled slot is entirely too small and that the slot is placed too low in the front.

Heat Losses from the Room. Since, in every case, the flow of heat was out of the room through the walls, floor and ceiling, and the only heat available for making up this heat loss was derived from the condensation of steam in the radiator, the only possible conclusion that can be drawn is that in the cases of the enclosures for which satisfactory air temperature conditions were main-

TABLE 2. COMPARISON OF STEAM CONDENSING CAPACITIES OF SIMILAR ENCLOSURES ON TWO DIFFERENT SIZED RADIATORS

Type of Enclosure	From Room Tests		From Bulletin 169	
	Enclosure No.	Relative Condensing Capacity Per Cent	Enclosure No.	Relative Condensing Capacity Per Cent
Bare radiator	..	100.0	..	100.0
Solid top, grilled front and ends, large free area	3	86.6	14	85.5
Solid top and ends, solid front with wide slot	10	82.7	11	83.5
Metal shield	1	88.4	Shield	91.9
Six-inch cloth cover	4	84.4	Cover	88.0

tained with less condensation than that obtained with the bare radiator, the reduced condensation was also accompanied by reduced heat losses from the room itself. For the satisfactory enclosure, the temperature at the ceiling was less than that for the bare radiator. Hence a reduction in heat loss through the ceiling resulted.

For the bare radiator, and for the enclosure having no protecting back, the temperature of the inside surface of the wall just back of the radiator was from 116 F to 121 F. For the satisfactory enclosure, this temperature was approximately 76 F. Accordingly the flow of heat through the wall back of the radiator was decreased by the use of the enclosure.

In the case of the bare radiator, the air that became heated by passing over the radiator rose and passed directly over the window. The velocity of this air was comparatively high and its temperature was considerably higher than the temperature at the breathing level. Furthermore, it was observed that there was no down current of cold air in contact with the glass under these conditions. When an enclosure was used the air passing over the radiator was deflected out into the room and did not pass directly over the window. Accordingly a greater loss of heat occurred by transmission through the glass in the case of the bare radiator than in that of the enclosed radiator.

The use of an enclosure undoubtedly reduced somewhat the amount of direct radiant heat received by the inside surfaces of the walls, thus reducing the surface temperature. A very small reduction in this temperature would not cause a noticeable reduction in the comfort of the occupants, but owing to the comparatively large area of the wall surfaces might represent a very appreciable reduction in heat loss through the walls as compared to the loss in the case of the bare radiator.

It may be noted that all of the reductions in heat losses are in themselves small, but that the sum of these reductions may easily be enough to account for the 0.73 lb of steam per hour, or approximately 700 Btu per hour, which represents the difference in condensing capacity between the bare radiator and the radiator enclosed with the best enclosure.

Relative Steam Condensing Capacity. In Engineering Experiment Station Bulletin No. 169 the relative steam condensing capacities of a radiator with various types of enclosures as compared with the steam condensing capacity of a 38-in., 20 section, 3-column bare radiator were reported. Since all of these tests were carried on in a large laboratory with the radiator placed near a warm wall and surrounded by air at a comparatively uniform temperature, no account could be taken of the actual heating effect produced by the different amounts of steam condensation. All conclusions in that bulletin were accordingly confined to the effect of the enclosures on steam condensing capacity alone.

In Table 2 a comparison of the relative steam condensing capacities reported in Bulletin No. 169 and those obtained from the present series of tests on enclosures of similar construction is given. The agreement between the two series of tests is remarkably close when it is considered that the two series were run under different conditions. Furthermore, the two radiators were of materially different size and the construction of the enclosures was not identical. Hence the present series of tests does not serve to alter any of the conclusions drawn in Bulletin No. 169 in regard to the effect of enclosures on steam condensing capacity, but does serve to present additional data on the relative heating effects to be expected in an actual room with cold outside wall and glass surfaces.

WARMING AND COOLING TESTS OF ROOMS

A few tests were conducted on the rate of cooling and the rate of heating of the air and walls of the test room. The temperatures of the wall and air were observed by means of thermocouples placed at the breathing level of the room. The location of the thermocouples is shown in Figs. 1 and 2.[7] In all of the tests an air movement of approximately ten miles per hour was maintained over the outer wall surface by means of three 16-in. oscillating fans located as shown in Figs. 1 and 2.

Heating of Room with Radiator Fitted with Enclosure. Test W-6, Fig. 18 shows the rate of heating of the room air and walls with the same radiator used in a previous test, except that a commercial enclosure (metal grille on front and ends) was used on the radiator. The temperatures at the start of the test were as follows:

Breathing line at center of room	= 23.6 F
Attic temperature	= 29.0 F
Basement space	= 38.0 F

The temperature of the cold room was about —7.0 F, as indicated by the average of the three thermocouples placed at the same height as the breathing level in the test rooms. An air couple was placed 6 in. from the inner wall surface of the test room at the level of the breathing line and opposite the inner wall surface couple. The locations of all these couples are shown in Figs. 1 and 2.

The shape of the temperature curves for this test are substantially the same as those for a similar test with a bare radiator. In both tests, the importance of allowing enough time for heating, and sufficient radiator surface for heating up periods, is well brought out by the curves of Fig. 18.

The temperature differences that were maintained after the tenth hour are approximately as follows:

Inside air 6 in. from wall to inside wall surface = 17.0 F.

Inside wall surface to studding space = 7½ F.

Studding space to outside wall surface = from 37.0 to 40.0 F.

Outside wall surface to outside air in cold room 4 F. The differences obtained were of the same magnitude as those observed in the previous test with a bare radiator.

Hourly steam condensate readings were recorded during this test and are also plotted on Fig. 18. The condensate curve is almost an exact reflection of the temperature curve for the room air temperature. The condensate weights obtained near the beginning of the test, when the inside air was below 40.0 F, were fully 25 per cent greater than those obtained toward the end of the test, when the room air temperature was about 62.0 F.

The enclosure used in this test produced temperature conditions at the breathing level in the room about the same as those in the test with the bare radiaor. The breathing level temperature with the enclosure was maintained at 62.5 F, while the bare radiator maintained 65.5 F, but the cold room in the former case was maintained at a temperature of —2 F, and in the latter case, at a temperature of —7 F, which readily accounts for the lower breathing level temperature in the room with the enclosure.

[7] In the following tests only the temperatures of the North wall of the West test room East end were plotted.

DISCUSSION

PRESIDENT WILLARD: In concluding this paper, before calling for discussion, I wish to make certain acknowledgments. This particular investigation, which, by the way, covered a period of about three years, is a cooperative investigation, supported not only by the University of Illinois, but also by the *Illinois Master Plumbers Association* and the *National Boiler & Radiator Manufacturers Association.* Both of these associations contributed funds for the necessary expenses of the investigation, the university furnishing laboratory space. Professor Kratz has presented to you a few of the results which will appear in the bulletin of the experiment station within the next few months. Many more tests will be made, and these rooms will be used in the future for the various investigations of different types of radiators. As you can see, the possibilities are almost endless, and with the plant that has been erected it will be possible to test radiators or heating devices in the actual environment found in practice, as referred to in the introduction of the paper and by the speaker.

E. H. LOCKWOOD (WRITTEN): I feel that our thanks are due President Willard and his associates for this valuable investigation, as well as for similar experiments that have preceded it.

My interest centers in the conclusions from Table 2, that relative condensation in bare and enclosed radiators was substantially the same, whether obtained in warm wall test rooms as in earlier experiments or in cold wall rooms of the present paper. The warm wall room is far simpler to operate, hence it is gratifying to get further evidence that its results will check well with the more elaborate and costly cold wall test room.

It will be equally important to know whether the absolute condensation figures for the two rooms will agree, assuming the same radiator to be tested in each. Such figures are not given in the paper, yet I believe that such comparisons may be fairly drawn. The coefficient of heat transmission has been worked out for the bare radiator as 1.87 Btu per hr per sq ft per deg Fahr. It seems to me that this coefficient should be recomputed, using the mean temperature of the living zone instead of the temperature at eye-height. If this is done the room temperature will be about 62 F, and the coefficient computed for room temperature of 70 F by the 1.3 power method will be 1.69, or almost exactly the coefficient given for this type of radiator in a warm wall test room. It is my impression that substantially identical results will be obtained from warm wall and cold wall test rooms, provided that the same temperature be maintained in the living zone in each.

It is certainly surprising to discover that a radiator enclosure will maintain the same living zone temperatures as the bare radiator, or better, with considerably less condensation of steam. It would be of interest to know how much of the steam saving was due to the presence of the protecting shield behind the radiator. Have any experiments been made with an enclosure consisting of a rear protecting plate only?

JOHN HOWATT (WRITTEN): The difficulty in artificially reproducing conditions met in actual practice when weather is one of the factors is illustrated in the set-up for the tests and experiments made on bare *versus* enclosed radiators at the University of Illinois. The cold room on two sides of the test room could maintain temperatures duplicating outdoor temperatures but the action of wind could not be duplicated by oscillating fans in the cold air space. It is a

well known fact that wind is as important a factor in the heating of a building in the winter season as is the outside temperature. Infiltration and leakage of air through walls and around doors and windows is what makes rooms in buildings of ordinary construction uncomfortable in cold, windy weather. The family sometimes has to move into the sheltered rooms to get away from the cold and drafts. Homes have flues from heating plants and fireplaces carrying air out from the interior of the home and increasing the amount of cold air coming in from the outside usually through infiltration or leakage. Would this condition make any material difference in the results obtained in the relative effectiveness of radiators bare or radiators enclosed? Enclosures certainly will change the relative ratios between the heat given off by radiation and convection so the air movement in the room would be a factor influencing the results.

I believe enclosure No. 10, Fig. 11, is of the general type found most often in homes, yet it has been shown by these tests to be the least effective of all. This is due to the restriction in the grille which checks the air movement through the radiator. In a paper by R. V. Frost, presented at the 1925 Annual Meeting of this Society, it was shown that the total amount of heat given off by a radiator could be very materially increased by the use of a properly designed enclosure. The writer had experience with enclosures similar to No. 10 a few years ago when he built a home heated with a hot-water heating system. The first winter the radiators were left bare and the house was kept comfortably heated when the regulator on the boiler was set to close boiler drafts when the water in the boiler reached 170 F. The appearance of the bare radiators did not please, so before the next winter enclosures similar to No. 10 were installed. That winter it was found necessary to raise the water temperature limit to 190 F to maintain equal comfort. No difference in the amount of fuel required could be detected.

I am convinced that, no matter what the tests of the relative efficiency of bare and enclosed radiators may show, the present-day boss in the home will have either concealed or enclosed radiators so the question is what type of enclosure. Even man, blind as he usually is, must admit that the ordinary bare radiator in the room has been an ugly thing. Enclosures are provided on radiators in the home upon demand of the women for three purposes:

1. To protect walls, curtains and ceilings from the dirt deposited by the column of heated air rising directly above the bare radiator
2. To conceal the usual ugliness of the bare radiator
3. To provide a means of humidification.

Those living in modern homes or first-class apartments are, therefore, demanding that the radiation be concealed or enclosed in metal artistically designed enclosures finished to match the wood work in the room. The results of tests such as those recorded in this paper should influence the design of such enclosures so they will satisfy the conscience of the heating and ventilating engineer by their effectiveness in producing room comfort, and at the same time satisfy the demand for an attractive-appearing heating unit in the rooms.

R. N. Trane (Written): This investigation marks an important step in house heating research and champions a new and more accurate standard for measuring the effectiveness of radiators. Instead of basing radiator performance on the amount of condensate, these investigators have chosen a more logi-

cal criterion in the mean room temperature from the breathing level to the floor which is the zone of useful heat. While it is true that other experiments have been run along these lines in the past it is not likely that the apparatus hitherto available has been as elaborately planned or as skillfully handled as the set-up in the University of Illinois.

The outstanding conclusions the authors have drawn from this investigation seem to be as follows:

1. One style of radiator may keep a room equally comfortable in the living zone, using less condensate than another.

2. All shields or covers used in this study have lowered the ceiling temperature in the test room below that observed with the bare radiator and delivered more effective heat per pound of condensate.

3. The same amount of comfort is secured with enclosure No. 3 as with the bare cast-iron radiator with a saving of fuel of approximately 13 per cent.

As long as all the enclosures which have been presented in this paper have been placed in the same room, scientifically arranged, and in identical conditions, it is very interesting to try to determine the efficiency of all the combinations of the bare radiator and the enclosures.

As long as the resulting temperatures of the various combinations vary, it seems logical to reduce the performance of each to a figure which equals the number of pounds of steam required per hour to produce a temperature difference of one degree between the inside and outside air, using the inside temperature either at the breathing line, in the living zone, or the average room temperature. In the case of the bare radiator we have the following three factors:

$$\text{Breathing Level: } \frac{5.44}{69.4 - (-1.1)} = 0.077 \text{ lb per hour per degree}$$

$$\text{Living Zone: } \frac{5.44}{62.0 - (-1.1)} = 0.086 \text{ lb per hour per degree}$$

$$\text{Average for Room: } \frac{5.44}{66.4 - (-1.1)} = 0.0806 \text{ lb per hour per degree}$$

The figures given in Table A showing this relationship between the various enclosures and showing the per cent saving of fuel as compared to the bare cast-iron radiator are obtained by this method.

TABLE A. RELATIONSHIP BETWEEN THE VARIOUS ENCLOSURES

Enclosure No.	Breathing level temp. °F.	Living zone temp. °F.	Average room temp. °F.	Net steam cond. per hour	Pounds steam condensed per deg. diff. per hour			Per cent saving		
					Brthg. level	Living zone	Room	Brthg. level	Living zone	Room
Bare	69.4	62.0	66.4	5.44	0.077	0 086	0.0806	0	0	0
3	69.5	63.6	66.6	4.71	0.067	0.073	0.070	13	15.1	13.2
10	66.2	57.8	62.3	4.50	0.0665	0.076	0.0705	13.0	11.6	12.6

From this table it will be seen that the statement by the authors on page 95 "This reduction (reduction of condensation effected by enclosures No. 10 and No. 11) can in no sense be regarded as an economy, however, because the enclosed radiator failed to heat the room," is not an evident conclusion.

The fact that enclosure No. 10 does not heat the room sufficiently is not of importance. The combination is too small, and could easily have been made larger by adding another section to the radiator.

Enclosure No. 3, without question, saves 13 per cent in steam to produce equivalent comfort, but enclosure No. 10 saves more (13.6 per cent) considering the breathing level temperature, slightly less (11.6 per cent as compared with 15.1 per cent) when considering mean living-zone temperature, and 12.6 per cent as compared to 13.2 per cent when considering the average room temperature. It does this, and produces a higher temperature difference between the floor and ceiling than any other arrangement, not excluding the bare radiator. In fact, enclosure No. 10 gives us 23 degrees difference whereas the cast-iron radiator gives us only 21 F. It is indeed a curious thing that enclosure No. 10 seems to do the impossible; that is, it increases this temperature difference between the floor and ceiling and seems to give more useful heat at a less expenditure of fuel.

These tests, if they are to be relied upon, and I can see no reason why they are not as nearly perfect as humanly possible to make them, indicate that there is something else that produces economy other than the difference between floor and ceiling temperatures.

The results which have been shown by enclosure No. 10 in producing a high difference in temperature between floor and ceiling could be expected from the construction of this enclosure. The free area of the grille is approximately 60 per cent and the radiator itself further obstructs this free area with the result that the flow of air through the enclosure is so restricted that the temperature of the air leaving the enclosure must be 15 to 20 deg higher than normal. Were this enclosure arranged with a somewhat better proportion of the opening for the discharge of air, and if the radiator in the enclosure were lower, at least below the bottom of the grille, surely a much more efficient combination would be produced because more air at a lower temperature would result. The relationship of floor and ceiling temperatures would also be greatly improved, and I would, therefore, recommend that these tests be continued with a great many more enclosures and different relationships of inlet and outlet air. The results from these tests will certainly be very much worth while. Certainly if four different enclosures selected at random have improved the economy of a cast-iron radiator there is great hope that many combinations will be found which will save a great deal more coal than enclosure No. 3, and that cast-iron radiators of themselves can be greatly improved.

Prof. A. P. Kratz (Written): In his written discussion, Mr. Trane has brought out some points that are certainly worthy of serious consideration. It is well to restate, however, that the original object and scope of the tests were to determine the effect of various existing commercial types of enclosures on the steam condensing capacity and distribution of heat in the room, when such enclosures were placed on a given size and type of a cast-iron steam radiator. The size of the bare radiator was just sufficient to maintain a breathing level temperature of practically 70 F when unenclosed, and the enclosures were selected so that the radiator practically filled the enclosure, extending to within an inch of the top.

A study of the results of these tests in regard to the effects produced must, therefore, be confined to the conditions existing in the room when this same radiator is used in all of the enclosures. What might happen if the size were increased or diminished is an interesting subject for speculation, but until further test data are obtained, any such figures must remain entirely in the realm of speculation. Furthermore, any attempts to correlate the performance of the

cast-iron radiator extending to within an inch of the top of the enclosure, with a low radiator or with a shallow copper-fin radiator placed at the bottom of the enclosure are at present rather futile. Some indication of what may happen to the condensation rate when high enclosures, giving marked chimney effect, are used is given in the results presented in Engineering Experiment Station Bulletin No. 169; but at present no tests have been run with such combinations in the test rooms, and the actual heating effects in the rooms are still problematical.

H. E. LONGWELL (WRITTEN): The authors have presented several definite facts which evidently have been determined with the utmost care and precision. The principal value of facts is that they enable us to develop a plausible theory which will account for them.

In heating a room, the characteristic most desired is uniformity of temperature at all points within the room. The degree of uniformity depends on the effectiveness of the diffusion of the current of heated air arising from the radiator. If this current is unobstructed, it naturally makes a short cut to the ceiling, giving us a hot zone at a level at which it is of no great service to us.

If we put a flat shelf above the radiator, the upward flow is obstructed, and the result is diffusion in a more or less horizontal direction into the zone in which we live. At the same time this obstruction to the flow of the heated air will reduce the amount of heat abstracted from the radiator, reducing proportionally the hourly rate of condensation.

The best performance may therefore be expected from that design of radiator shield or enclosure which gives the most effective diffusion of the heated air, with the minimum of obstruction to its flow.

Of the several types tested Fig. 10 exhibits these characteristics in the highest degree. It has a solid back which subjects all of the circulating air to the directional control of the grilles. The grilles are noticeably open and large, consequently offering a minimum of obstruction to the circulation. There are grilles in the ends of the enclosure as well as in the front, which not only lessen still further the obstruction to the circulation, but also make the diffusion more or less radial through an arc of 180 deg. The bottom of the grille is low, so that there is considerable diffusion near the level of the floor.

Fig. 11, on the other hand, has no back, and consequently the directional effect of the grille is materially weakened. The area of the grille is considerably less than half of the area of the grille in Fig. 10. On this account the air flow would be reduced and it is not surprising to find that the hourly condensation, as shown by the test, is less. Having no grilles on the ends, all of the diffusion is unidirectional, and therefore less extensive than is the case with Fig. 10. Finally, the bottom of the grille appears to be about 18 in. above the floor, and therefore we cannot expect much help from diffusion below that level.

It might be instructive if the experiment with this type of enclosure could be repeated, using a larger radiator so that at its reduced efficiency it would still be able to heat the room, at the breathing point, to 70 F.

It might also be worth while to make another experiment with the enclosure Fig. 10, elevating it on blocks 1 in. high, in order to determine whether its performance is affected by a moderate increase in the area of the air inlet.

In the case of Fig. 12 we have the solid back shield, and no obstruction by grilles at either the front or the ends of the radiator. We also have a greater

hourly condensation than with enclosure Fig. 10, and yet the performance is somewhat inferior. It is possible that on account of the shortness of the top shelf, there is not sufficient overhang to deflect the upward air currents caused by the heat emitted from the ends of the radiator. In a radiator having only a few sections, this is a considerable proportion of the total heat. In the case of the present experimental radiator, it would probably be close to 25 per cent. Assuming that any such proportion of the heated air were shunted directly to the ceiling, it is not difficult to believe that the effectiveness of the shield as a diffuser would suffer materially. If the experiment were repeated using a shield of this type, having a top as long as that of enclosure Fig. 10, and the back extending the entire length of the top, I fancy we might get results fully as good, if not a little better, than were obtained with Fig. 10. Of course, irrespective of how excellent the performance might be the aesthetic atrocity of the structure would outweigh its merits hopelessly.

The general diffusive effect—which is the real essence of so-called comfortable heating—can be obtained to a very considerable degree, without recourse to shields or enclosures, but merely by exercising good judgment in the selection of radiators.

A tall radiator made up of a few sections heats a comparatively small quantity of air, per unit of time, to a comparatively high temperature. Due to the high temperature the velocity of the air current is so rapid that there is but scant opportunity for diffusion before the ceiling is reached. Consequently we have a very steep temperature gradient from the ceiling to the floor. If we maintain the same footage, but use shorter sections and more of them, the radiator heats a larger mass of air to a lower temperature. While it transmits the same quantity of heat per unit of time, the velocity of the air current is reduced, diffusion is increased, and the temperature gradient between ceiling and floor becomes less steep. If we use a still greater number of sections with fewer tubes, the diffusive effect is still further increased. If we use hot water at 150 F as a heating medium instead of steam at 215 F, and increase the number of sections in the radiator 60 per cent, a still greater degree of uniformity of temperature will result.

I should be glad if the authors could find occasion to make another test with a bare radiator, 3-tube, 20 in. high and 12 sections, under exactly the same conditions as prevailed in the test of the 5-tube, 26 in., 6-section radiator. I would hopefully expect improved comfort conditions not only by reason of better diffusion, but also on account of a marked increase in the percentage of radiant heat. However, even though it should fail my expectations from a purely utilitarian standpoint, I am confident that in a small room answering to the description of these tests rooms, a radiator of the proposed dimensions would at least be infinitely more satisfying to one's sense of artistic fitness—which, in these modernistic days, is not to be overlooked.

C. H. B. Hotchkiss (Written): Many of us have been waiting for some time for the first report of results from this new Illinois plant. Now that they are here, it is difficult to understand how anyone interested in the problem of testing radiators can be disappointed, or can be anything but delighted. From the description of the plant given in the paper it is so evident that both the design and the construction are well thought out that no comment is necessary. Now that the paper reveals that the operating results have been so satisfactory, we should all feel very well pleased.

The operation of the plant is fully described. However, there is one point

which is not entirely clear. Direct expansion coils are used in the cold room, and the low temperature is apparently maintained by causing air to move over those coils by the fan. It appears that only three temperature measurements are made in the cold room. One of these must be the one used for controlling the action of the automatic expansion valve mentioned. The plan shows it just north of the west test room. The plan shows no points of temperature measurement near either the floor or ceiling in the cold room, nor in the passageway between the two test rooms. It is evident from the method of cooling that there must be some fairly considerable variation in temperature between the air at the floor and at the ceiling and in the passageway. As it seems quite likely that such observations must have been made at some time during the testing, it might have been desirable to have included such data in the paper. Its inclusion would have assisted greatly in forming an idea as to the average temperature in the cold room, as well as to have made clearer just how effective the oscillating fans are.

The paper brings to light one item of information which should not be permitted to pass without calling attention to it specifically. It has been apparent to every one concerned that the temperature of the wall surface direcly behind an unshielded direct radiator must be high. It is one of the unexplained vagaries of test work why no one has ever before seen fit to attempt to measure, and publish the amount of, this temperature.

The paper states that there are four points of importance to be considered in drawing comparisons and conclusions as to the performance of radiators and also gives five factors of performance. As one of the major points involved in all testing of radiators under conditions of use revolves around an agreement on these factors, it seems evident that the major discussion must hinge on them.

The very fact that the authors recognize and admit that condensation is not itself a full measure of radiator performance is a step in advance, in that it shows an attempt at the analysis of those factors that cause satisfctory and desirable distribution of heat. It is also notable that the authors attempt to set up no one point of measurement of temperature as a criterion as has so commonly been done in previously reported tests of radiators in test rooms.

Certainly the statements as actually set forth in words by the authors are open to question. Even if one agrees that the list of performance factors is reasonably complete, or complete enough for practical purposes, there are questions involved which will not down. For instance, the factor (4) as set forth by the authors is "if the mean temperature in the living zone was raised." This implies that the temperature at the top of this zone is being kept substantially steady as required by factor (1), and that the floor temperature is being kept steady or is raised as required by factor (3). How can the factor (4) be fully reconciled with the statements of factors (1) and (3)? Do not the factors (1) and (3) cover the requirement of factor (4) so long as temperatures are recorded only at the center of the test room? If, however, measurements of temperature were made at points distributed over the whole room then it would seem that the factor (4) is valid and very meaningful.

This brings up a point of some interest and one which it seems to me is worthy of thought. In our testing plant we have placed thermal couples not only at various heights in the center of the room but also at the same approximate heights near the corners. We have shielded these couples from direct radiation and when direct radiators are used to heat the test room the couples show

temperatures quite notably different from those in the center of the room. Those in the two corners near the hot radiator show temperatures varying a good deal from those in the corners farthest from the radiator. As we have not yet used any devices except direct radiators in the room it is not possible to predict what temperatures will be shown by them. However, it seems quite possible, and even probable, that some considerable differences might be shown, especially in those forms of heaters which direct the warmed air outward across the heated room. In such cases the variation between the living-zone temperatures at the corners of the room and at the center should be appreciably lessened. Why should such a device not be credited in some way for whatever beneficial effects it is able to produce because of this better temperature distribution in the living zone? Isn't it just as desirable to have a 70 deg five-foot temperature near the corners of the room as at the middle, for, don't occupants use other parts of the room as well as the center? Why, then, isn't it just as logical to include some such factor in the list as to include those already there?

This omission seems to constitute about the only one in an otherwise very well thought-out testing setup. It should be of interest to hear why more temperature measurements in the rooms were not made since their inclusion would have meant but little additional effort. It would seem, too, that their inclusion would have been desirable for the sake of completeness.

Thus it is fairly evident that while the paper presents certain factors on which radiator performance may be judged it is open to question as to whether or not the list is fully complete. Instead, does not the list merely show those factors concerning which data were collected in those particular tests?

It is also apparent that while the list sets forth certain factors of performance, no attempt is made to assign a numerical value to the relative importance of each. It is thus still in the stage of being a rather rough scale which does not permit of separating out marginal cases by its use. However, it should not be criticized from this standpoint, but rather we should all be glad that so much headway has been made regardless of opinions as to details.

Another point also occurs. The authors state that with both the bare and shielded radiators and a wind velocity of approximately 10 mph certain results were obtained as to temperature distribution. It seems quite clear that the reported tests were all conducted with the air moving at this velocity. I should like to have the authors confirm this conclusion.

It seems to me that the data showing the rate of heating are of especial interest and importance. In the first place this seems to mark the first appearance of such data as obtained by laboratory methods similar to those used. Other curves of the heating or cooling of building wall constructions have been obtained by calculation.

However, the fact that the curves present a new kind of data is not the principal reason for their interest. It is apparent that since data of these kind can be obtained from this plant that it will be possible to conduct heat transmission tests under somewhat different conditions than have been used heretofore. Moreover, it should be possible by the use of the plant to settle some rather vexing questions regarding our existing notions of heat transmission—namely, on the rate of temperature penetration of building materials. I have no idea what is intended in the way of a testing program at the plant but it certainly holds the possibility of changing our methods of computation as profoundly as

did the use of coefficients change the methods of estimating in use before their introduction.

KONRAD MEIER (WRITTEN): The experiments at the University of Illinois bring out certain facts about direct heating, that had long been known, but did not receive due consideration, partly because of insufficient demonstration. Even now it seems doubtful, whether they will be of practical value, unless conclusions are fully drawn and stated in a manner to compel attention and reform.

The principal point, which the tests were to settle, was the comparative room heating effect of bare, shielded and enclosed radiation, which would show, whether these familiar devices are advantageous or not. Comfort and health being the first considerations, the results are properly expressed by the temperature gradient, although the time, within which say 65 F are reached near the floor, would also be of moment in this respect. With 69.5 F at the breathing level and about 60 F at knee-height and after a long period of heating up, the difference between floor and ceiling for the bare 5-tube radiator was found to be 21.8 F. In a room of only 9 ft height and neutralized transmission through floor and ceiling, this is certainly not a good showing in regard to the placing of the heat. It explains the lack of comfort and indicates the reason for the customary, but in reality, excessive standard of room temperature. With the deflecting shelf and the enclosure having grilles on three sides, which alone gave some improvement, the temperature differences were still 20 and 19 F respectively and the increase near the floor was barely perceptible. Both arrangements have a top shelf deflecting the warm air not only towards the room, but allowing also escape sideways. Evidently, it is the latter feature, not present in any of the other devices, which causes at least a partial spreading of the warm air under the windows and accounts for the slightly better showing. The short unit tested is not favorable in that sense, particularly as placed, since the chilled air from the windows can reach the floor for nearly the entire width, while a hot column of air is allowed to rise straight to the ceiling. It would be easy to improve on this performance by appropriate selection and disposition alone. As might have been expected, the other forms of coverings, including the typical enclosure No. 10, were still less efficient, mainly because of the absence of radiant effect, which fact appeared to offset the benefit of deflection, as the parallel curves would indicate.

Thus, although we are dealing with a radiator located under the window, its heating effect appears to be little better than that of surfaces on inside walls. However, as a general conclusion, this would be misleading, even though the arrangement adopted for the tests may be typical. The fact is, it does not represent good practice, as the preliminaries must have shown at once. Hence, while it was decidedly useful to demonstrate its characteristic faults, the question may well be raised, whether such an arrangement should have been used as a basis for comparison. In any event, it was shown, that its disadvantages are not overcome to any extent by ordinary deflectors and even less as a rule by coverings intended primarily to improve appearance.

The principal factor governing room heating efficiency is location, since the best results are not attainable with any kind of surface, unless properly placed. But the investigation also makes it clear, that application of the heat under the window, in itself, does not assure the desired effect. Whenever radiators are placed in that way or another, they should be of a type suited to the location. In this case it means low and shallow patterns, which are also the least obtru-

sive. Such types could be produced at about the cost of wall radiation and would be advantageous, not only in giving the best radiant effect within the lower zone, where not objectionable, but also in spreading the warm air, so it will mix with the colder currents and prevent a cold floor. The temperature difference between floor and ceiling will then practically disappear and it takes less time to reach the desired comfortable condition. It should go without saying, that the heat output must be regulated according to the weather and intense local effects avoided. Thus, high efficiency may be readily secured by judicious selection and application of the surfaces and without resorting to any deflectors and enclosures. In fact, these will then only impede the proper action. They will add nothing whatever to comfort. By gathering any drying dust, which vitiates the room air, they are more likely to become a source of irritation.

We are all aware, that enclosures are in reality concessions made to clients who are not willing to accept the imperfections of current practice. Rightly or wrongly, they are designed for architectural effect rather than for efficiency and very little could be accomplished by their improvement. Deflectors, on the other hand, are devices intended to correct certain faults in placing the heat, which they may do to a limited extent, but at the expense of appearance. What we need today, is an all-around satisfactory solution of the various problems involved: Adaptability to space and location combined with finished appearance, thermal efficiency and hygienic quality. The investigation might be of great value, if kept up with that end in view. And it would show that all the reasonable requirements can be met with plain exposed cast-iron radiation, when available in the cleanest, most efficient and least cumbersome form, which should induce proper placing and lend itself without further aid to neatest installation.

PROFESSOR KRATZ: Most of the questions that have been raised, I think, have been answered in the presentation of the paper, and a large part of the discussion is merely pointing out and corroborating the results obtained in the tests.

Professor Lockwood asked something about the agreement of the test results between the two rooms.

In every case, we ran check tests on the two rooms, each combination was tested in each one of the two rooms and sometimes a number of tests were run, and in every case we found very close agreement between the results obtained in the two rooms. I have not the exact percentage, but I would say it was somewhere between, within one and two per cent agreement, the results of the two rooms.

Since running some of these tests, the same radiator has also been placed against the warm wall and tested under the same conditions as the report in Bulletin 169. We obtained very close agreement between the heat transmission coefficients determined by the tests in the warm and the cold room. I think that answers Professor Lockwood's question in that respect.

In regard to the protecting plate: undoubtedly a large part of the heat was saved by the protecting plate at the back of the radiator. The temperature was reduced from about 121 F to 76 F by the use of the plates. Calculation, I think, indicated that in the neighborhood of one-fourth of the heat saved was saved through that source.

Mr. Howatt asked something about the infiltration. Undoubtedly, infiltration would affect results to some extent. Our calculations indicate that there was approximately one-fourth of an air change per hour in the room. The fact that

we had two windows on one side and a door on the other side of the room helped to get some infiltration. No weatherstripping or any unusual conditions of tightness were observed on the windows and door; so we did get some infiltration, possibly not as much as you would expect with a high wind and loose construction. Therefore the infiltration effect has not been entirely disregarded. The infiltration, by the way, was the same in all cases.

Mr. Longwell asks what would happen if a moderate increase in the area of the air inlet were made. We did not make any of those tests in this room, but the results in Bulletin 169 indicate that a moderate increase in the inlet area would also be accompanied by increase in the steam condensation and probably the same results would maintain here, since we found a very close correlation between the results in the room and results obtained against the warm wall, so far as condensation was concerned; that is, the same thing done to the radiator in the room as done to the radiator against the warm wall produced about the same percentage increase or decrease in condensation as the case may be.

Mr. Longwell suggests a number of tests. We are not by any means through with this room, and we will be glad to give consideration to additional tests of the type that have been suggested.

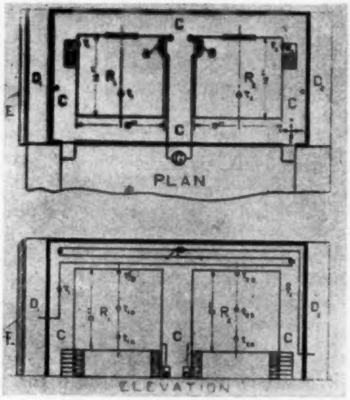
PRESIDENT WILLARD: I would like to call specifically on Dr. Brabbée since he has similar equipment and has done similar work in the field.

DR. C. W. BRABBÉE: You remember that, at the last Annual Meeting, I discussed our radiator twin test rooms, as shown in Fig. 1a, and demonstrated that the judging of radiators by their condensation only gives entirely different results from those obtained if the comfort effect for human beings is considered. At that time the objection was made that our results were an expression of the characteristics of the rooms and not of the radiators.

The same objection would hold good for Professor Willard's test arrangement, because in this case the radiators are also investigated in rooms of certain sizes and arrangements. We have made some more investigations in this matter and it might be interesting to show what results have been obtained.

Two radiators *B* and *C* have been tested in our twin test-room arrangement against the standard Peerless, and the results obtained tabulated. Radiator *B*, which is a flat shallow one, 17 in. high, has an actual heating surface of 8.35 sq ft per section, the rating by condensation is 8.04 sq ft per section but the rating by useful heat is 9.55 sq ft per section. Radiator *C*, a 5-tube heater, 19½ in. high, has an actual heating surface of 2.05 sq ft per section, a rating by condensation of 2.1 sq ft per section, and a rating by useful heat of 2.37 sq ft per section; these figures obtained, as stated from investigations in our twin test rooms. You see that with Radiator *B* the useful heat rating is 19 per cent higher than the condensation rating, whereas Radiator *C* has a 13 per cent higher useful heat rating than according to condensation. This result would indicate that the flat heater *B* will give, with less condensation, more useful heat, the total difference being expected to amount to about 6 per cent.

Now, both these Radiators *B* and *C* were removed from our twin test rooms and installed in Rooms Nos. 1 and 2 of Fig. 3a, both underneath the windows in rooms which have entirely different physical dimensions, and also entirely different outside walls and outside windows. However, both rooms are absolutely alike except in their location against the sun. Therefore, the tests were made at night after cloudy days. From Fig. 3a we see that in a state of equilibrium



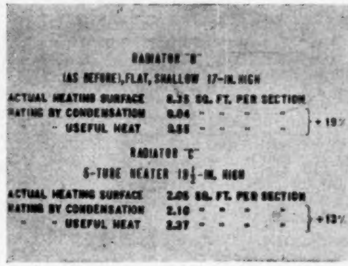

RADIATOR "B"
(AS BEFORE), FLAT, SHALLOW 17-IN. HIGH
ACTUAL HEATING SURFACE [illegible].35 SQ. FT. PER SECTION
RATING BY CONDENSATION [illegible].04 " " " "
" " USEFUL HEAT [illegible].55 " " " " } + 19%
RADIATOR "C"
5-TUBE HEATER 19½-IN. HIGH
ACTUAL HEATING SURFACE 2.06 SQ. FT. PER SECTION
RATING BY CONDENSATION 2.10 " " " "
" " USEFUL HEAT 2.37 " " " " } + 13%

Fig. 2a. Results of Testing Radiators *B* and *C* in Twin Rooms

←Fig. 1a. Radiator Twin Test Rooms

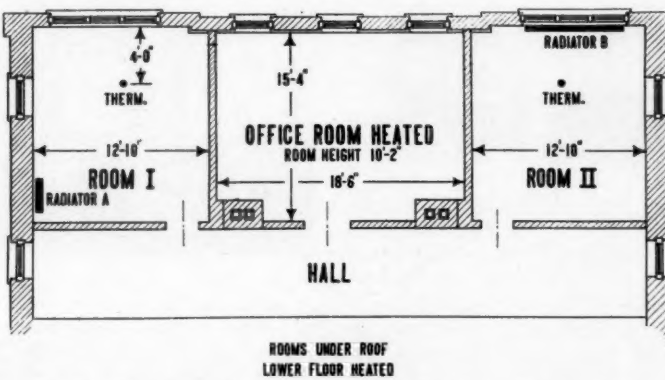


Fig. 3a. Radiators *A* and *B* Shown in Practical Installation

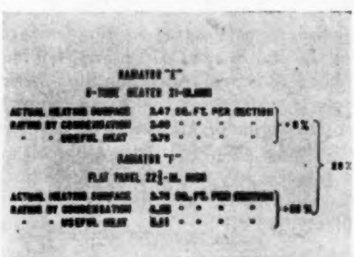
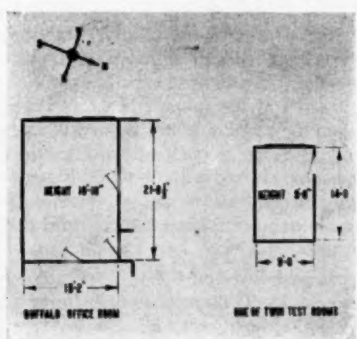

RADIATOR "E"
ACTUAL HEATING SURFACE 3.47 SQ. FT. PER SECTION
RADIATOR "F"

Fig. 4a. Results of Testing Radiators *E* and *F* in Twin Rooms

Fig. 5a. Rooms Used to Compare Heating Capacity of Radiators →



Radiator *B* has a condensation of only 7.28 lb per hour, whereas, at the same time, Radiator *C* condenses 8.23 lb per hour which is 13 per cent more than the condensation of Radiator *B*. However, the knee-height temperature curve indicating the feeling of comfort is 2 F, with radiator *B*, higher than with Radiator *C*, which, figured by our standard method of calculation, gives a 7 per cent higher useful heat rating of Radiator *B* against C. In other words, with a 12 per cent less condensation, 7 per cent more useful heat with the flat radiator *B* is obtained, the latter therefore being 19 per cent more valuable than Radiator *C*, whereas an advantage of only 6 per cent was expected. We see that the practical experiences, namely, the preference of the flat Radiator *B* over *C* is in the same direction as obtained in our twin test rooms, only more favorable for the flat heater, which is explained by the fact that our method of test includes a considerable amount of reserve as advisable for the introduction of a new product.

Another example that the tests made in our twin test rooms are in the right direction is shown in Fig. 4a. We investigated in our twin rooms a Radiator *E*,

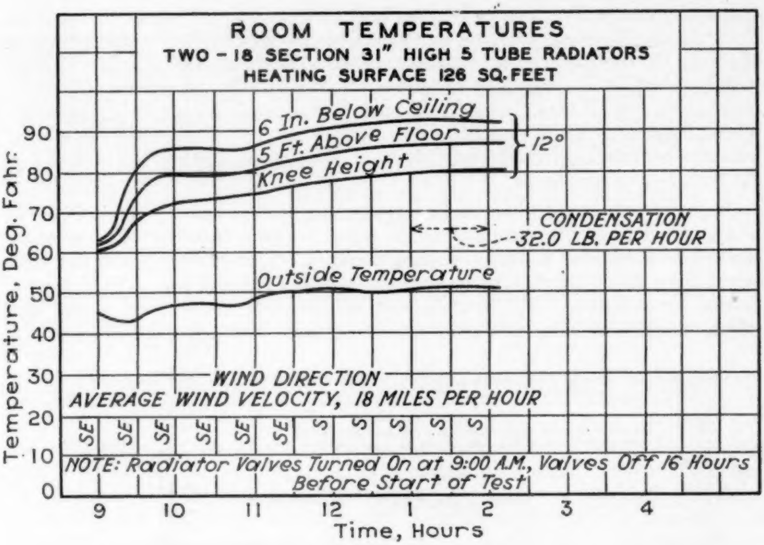
FIG. 6a. TUBULAR HEATER INSTALLATION

FIG. 7a. FLAT PANEL INSTALLATION→

5-tube, 30 in. high, and a Radiator *F*, flat and panel-like, 22¾ in. high, and found that figuring by useful heat, the latter could be rated 28 per cent more favorable than the tubular heater *E*. We then installed these radiators in a room, 15 ft 2 in. x 21 ft 8½ in. x 10 ft 10 in. high, in one of our offices in Buffalo, with two outside exposed walls, in each one two windows, all these details greatly different from our test rooms.

Fig. 6a shows the tubular heater, of which 126 sq ft actual heating surface were installed and which were replaced later by flat heaters (Fig. 7a), having together only 90 sq ft actual heating surface, *i. e.*, 28 per cent less than the former 120 sq ft of the tubular heater. Fig. 8a shows the practical tests. On the day when the tubular heater was investigated the outside temperature was between 45 and 50 F, the wind blew from south-southeast at 18 mph, and the heating up of the room started with about 60 F room temperature. After four hours a practical state of equilibrium was obtained, during which a condensation of 32 lb per hour was measured with a temperature at knee height of 80 F.

Then these 126 sq ft tubular radiators were removed and 90 sq ft flat heaters were installed and the test repeated. The outside temperature in this case was around 15 F, therefore much lower than on the previous test. The wind blew from the northwest with a velocity of 17 mph, therefore under much more unfavorable conditions than with the test of the tubular heaters. The heating up



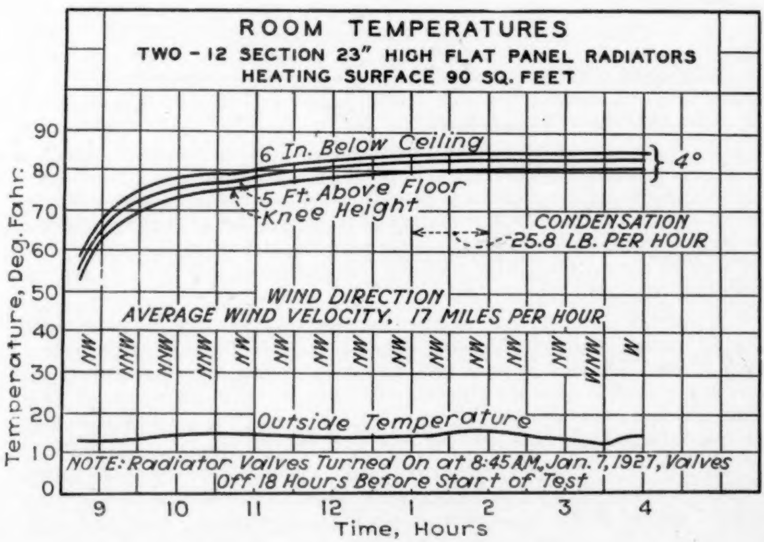


Fig. 8a. Radiator Test Results

of the room started with about 52 F at knee height, also more unfavorable than in the former case, and yet, after four hours, the temperature at knee height was again 80 F, whereby the condensation was only 25 lb per hour. You see that with a reduction in actual heating surface of 28 per cent, and under much more unfavorable conditions all around, the same temperature at knee height was obtained, indicating that the results of our tests in the twin test-room arrangement were again in the right direction but with a considerable reserve in favor of the flat heater as desired by us.

One more example of what practical results can be expected is given in Fig. 9a, which shows three flat radiators out of a total of four installed. The room is exposed to north and east, has wooden windows, weatherstripped, and the heat calculations called for 114 sq ft standard radiation figures for +10 F lowest outside temperature. We installed four of these flat heaters and forgot one which should have been under the north window. The actual heating surface

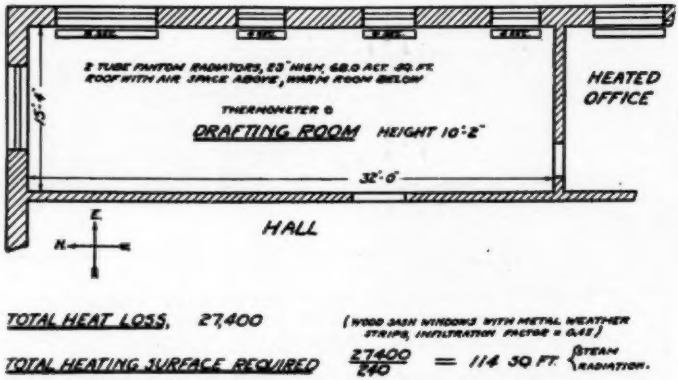


FIG. 9a. SPECIAL TEST INSTALLATION OF PANEL HEATERS

set-up is only 68 sq ft equivalent to 85 sq ft, rated at useful heat, with one radiator of seven sections or 29 sq ft heating surface shy. Yet on January 14 when the outside temperature was only 7 F, the knee-height temperature was 69 F at 9 o'clock in the morning, with a ceiling temperature of only 69½ F, and the conditions were very comfortable.

It is interesting to note that, as on all previous occasions, the flat heater with a large amount of direct radiation in the lower part of the room provides a much smaller difference between knee-height and ceiling temperatures. In the tests in our Buffalo office the difference was only 4 F with the panel and 12 F with the tubular heater. We may infer from these practical investigations that the results obtained in these twin test rooms are not only functions of the rooms but will hold good for practical installations if certain precautions are observed.

Before I close, it might be interesting to show twin test rooms (Fig. 10a) which I erected at the University of Berlin, Charlottenburg, in 1920 for investigation of tile stoves in regard to useful heat. In 1922 we erected twin test rooms for testing radiators as seen in Fig. 11a, whereby we again strove to

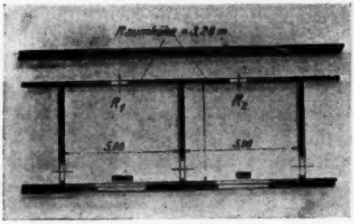


FIG. 10a. TWIN TEST ROOMS AT UNIVERSITY OF BERLIN

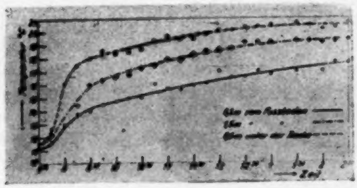


FIG. 11a. TEMPERATURE DISTRIBUTION IN ROOM

reach for the goal of determining the useful heat output of radiators and studied the temperature distribution in the room at knee height, eye height and ceiling height. (Fig. 11a.)

In very recent European papers with most elaborate investigations about radiators, everything is based on condensation only and not a single thought is given to useful heat output, but in America, gentlemen, we have again dug out the old pet idea. The American Radiator Co. gave the means for the investigation and four years ago at the Annual Meeting in Buffalo I had the honor to present to you the problem of useful heat. Now for the first time an official voice is heard and in principle our ideas have been adopted: not that radiator is best, which gives maximum condensation, but the one which provides maximum human comfort with minimum condensation and therefore minimum fuel expense.

We convey our heartiest congratulations to Professor Willard and his co-workers for this splendid contribution to the further development of our art. New ideas for testing, designing and installing of radiators have been officially set free and a new chapter in the history of heating has just been written by the University of Illinois, a chapter which may well be entitled—"The Era of Useful Heat."

PRESIDENT WILLARD: The authors of this paper express their sincere appreciation of the very complimentary remarks of Dr. Brabbée. He is a pioneer in this field of knowledge, carrying the frontiers of knowledge farther every day. And may there be more power and long life to him!

I will now entertain a brief verbal discussion of the paper. I assume there has been a great deal of material already presented which has answered questions which may have occurred to you; but if not, you may discuss the paper briefly.

N. W. DOWNES: Two of the enclosures had humidifying pans. Was water used in the pans during the test?

PROFESSOR KRATZ: All of the enclosures had humidifying pans, but no water was used in any of them in any of the tests, largely because we wanted to compare the enclosures under exactly the same conditions. Humidity readings in the rooms showed relative humidities varying from 15 per cent up to 25 per cent, and that is practically the same humidity that obtains in the rooms in ordinary houses. We felt that we were not running under conditions in any way outside of the ordinary. The humidity would not affect the distribution.

F. D. MENSING: What were the temperatures in what we might call the basement space? What were the temperatures in what we might call the attic space, and why were those temperatures maintained?

PROFESSOR KRATZ: The temperature in the attic space was 62 F. The temperature in the basement space was always maintained about 2 degrees higher than the temperature 3 in. above the floor; that is, if the temperature above the floor was 56, the air temperature in the basement space was 58. In other words, we wanted practically balanced heat flow through the floor. You asked why they were maintained at those values. Because preliminary tests showed this: that they were about the values we had to have in order to keep from reducing the radiation below 21 sq ft. We felt if the radiation was reduced below 21 sq ft we would be getting conditions that were hardly representative. If we had a completely cold floor and ceiling, we found it would require more radiation, and would overload our refrigerating plant.

R. C. BOLSINGER: I would like to ask Professor Kratz about the design of the enclosure at the top. Was it flared or square top on the inside?

PROFESSOR KRATZ: All square top with the humidifying pan in. You refer to the design under it?

MR. BOLSINGER: Under the top.

PROFESSOR KRATZ: All square.

MR. BOLSINGER: Was your test started with a cold radiator or was the radiator heated?

PROFESSOR KRATZ: I forgot to mention that. The condensation was not measured until the temperature on the inside surface of the room showed that the room had been in equilibrium for anywhere from a half to two hours. No data were taken when the rooms were not in equilibrium, and our criterion for equilibrium was constancy for the inside surface temperatures of the wall.

E. K. CAMPBELL: Professor Kratz stated that in general the shield which interfered least with the air movement over the radiator produced the best results. I am wondering if a more correct statement would not be that the shield which increased the air movement over the radiator the most produced the best results, and the shield which decreased the air movement the most produced the poorest result.

We know in heating large rooms with higher ceilings by air movement under fan conditions that a large volume of air heated to a comparatively low temperature will in all cases decrease the difference in temperature between floor and ceiling. Isn't that the real explanation of at least part of the beneficial results obtained from the shields which increased the air movement and also from the long radiator which Dr. Brabbée uses, the flat radiator, which increases the air movement in proportion to the surface of the radiator?

PROFESSOR KRATZ: Mr. Campbell's statement is just a corollary of the statement the way it was made; that is, the shield which offered the least resistance, of course, would give you the maximum increase in the amount of air going over the radiator, provided that there is an increase.

H. M. HART: Time is short and I am not going to say anything about the past, but about this test in which I am interested.

May I suggest that this paper when published contain a detailed cross-section of the top of the radiator. I am inclined to think that that construction has a

great deal to do with the effect and the distance from the top of the radiator to the bottom of the apron which permits the air to flow out from the enclosure. I think that has a great effect and that such details would add somewhat to the value of the paper.

M. T. Clow: I would like to inquire if the temperatures of the air delivered from the radiators were obtained. Such temperatures would be interesting as an indication of the useful heat rating of radiators—that is, a low radiator has a higher useful heat rating than a high one, because the temperature of the air delivered by the low radiator is lower than that of a high radiator. On the other hand, the useful heat rating of an enclosed radiator would be greater than that of an unenclosed radiator due to the horizontal direction of the air delivery, in spite of the fact that the temperature of the air might be higher for the enclosed radiator.

Professor Kratz: Just before coming to this meeting some temperatures of the kind mentioned were obtained, but there is not very much difference between the temperature of the air delivered above the bare radiator and that delivered from the shield. We have not had a chance to study these data very carefully, but we expect to take more data and go into that question in more detail.

In reference to a suggestion that Mr. Hart made, the final report will contain a drawing such as he indicates. And I might say that practically all of the enclosures had a distance of about one inch between the bottom of the water pan and the top of the radiator. In getting the enclosures, that distance was specified, and I think it was held to a distance of three-quarters to an inch and a quarter.

Mr. Hart: Was not the opening in the front of the enclosure below the bottom level of the pan?

Professor Kratz: In some cases it was slightly below it and, I think, in some cases practically on the level.

Percy Nicholls: I would like to ask the author whether the temperatures he gives at breathing height, knee height, and so forth, are the average temperatures at a cross section of the room parallel to the radiator, or the temperatures opposite the center of the radiator, and whether an average over the cross section might not be affected by having openings on the side of the covers for the radiator, as well as on the front.

I would also like to suggest that while such excellent and very thorough work is being done, it might be carried one step farther by measuring the effect on the comfort of the room as produced by the temperature of the air and the heat which the body receives by radiation. It seems to me an omission that work as thorough as this should be done without gathering such data as well. I know it is a rather difficult job, but I think that instruments could be devised whereby you could get the true comfort that the body would receive from the air temperature and from the radiation, both due to the cold walls and due to the hot exposed surface with the bare radiator as compared with the covered radiator where there would not be much radiation of the enclosure.

Professor Kratz: In reference to Mr. Nicholls' first question, I assume that he refers to what is the effect of radiation on the thermocouples used to measure the temperature. The thermocouples used were No. 22, B. & S. gage wire. Three of these couples were placed at the center line of the room and one of

the couples was shielded by an ordinary radiation shield, one was left bare, and one couple was wrapped with some woolen string in order to find out whether there was any appreciable effect of radiation. The result was that the bare couple and the shielded couple read the same. The couple that was wound with woolen string read slightly lower, about half a degree, than the other two. In other words, the radiation which was being received by the couples was not the hot radiation from the radiator, but in the aggregate the cold walls were affecting it more than the radiator, but the correction never exceeded more than half a degree and we accepted the readings of the bare couples as being representative.

All readings were taken on the axis running through the center of the room.

In reference to the use of the kata-thermometers to obtain the actual comfort conditions, we found that entering the rooms disturbed conditions to such an extent that during a test we never opened the door; and as far as I am able to determine, the readings of the kata-thermometer must be made with the operators very close to the thermometer. We were afraid it would disturb conditions too much to use the kata-thermometer, but we expect to try that out in the near future.

L. A. HARDING: As I understand it, these tests were not run within the limits of the comfort zone. You admitted it felt cold in the room, and I believe you carried a relative humidity of approximately 15 per cent.

PROFESSOR KRATZ: Fifteen to twenty-five.

MR. HARDING: That is not within the comfort zone.

PROFESSOR KRATZ: No.

MR. HARDING: The only suggestion I wish to make, in reference to further tests along this line, as comfort is a feature, is that the combinations of temperature and relative humidity be kept within the comfort zone.

Your tests indicated a saving in radiation of approximately 15 or 16 per cent. I do not believe that was brought out very forcibly. Dr. Brabbée's tests with this special radiator indicate a saving of about 33 1/3 per cent. Of course, a different type radiator was employed. These savings are of such great amount that we all look forward to the time when engineers are willing to accept the figures to be applied in actual practice.

This investigation is a real step in the right direction.

No. 822

EXPERIMENTS ON THE EFFECT OF SURFACE PAINTS ON RADIATOR PERFORMANCE

By C. H. FESSENDEN[1] AND AXEL MARIN[2], ANN ARBOR, MICH.
NON-MEMBERS

Purpose

THE purpose of this paper is to describe the method used and present the results obtained during some recent investigations made at the University of Michigan to determine the effect of certain paints on the amount of heat emitted by a radiator.

The topic is not new. Prof. John R. Allen presented a paper[3] about eighteen years ago in which it was reported that paints having a flake metal base when applied to a cast-iron rectangle reduced the heat emission of the radiator as much as 25 per cent. Although the statement has frequently been made that the form and distribution of the surface of the radiator would influence such quantitative data, the erroneous impression seems to be rather widely spread that aluminum paint reduces the effectiveness of *any* radiator by about 25 per cent. A later paper[4] by the same author stated that a two-column, 38-in. radiator, 10 sections long, when painted with aluminum bronze transmitted 200 Btu per sq ft per hour as compared with 240 for the bare cast-iron, a reduction of 16.7 per cent. He further stated that 62 per cent of the 240 Btu emitted by such a radiator is transferred by convection and the remainder by radiation and that different paints influence only the amount of heat radiated and do not affect the amount transferred by convection.

Some experiments to determine how much of the total heat emitted by a radiator is dissipated by radiation, reported[5] by Allen and Rowley in 1920, include data which show the effects of paints. A two-column, 13 section, 38-in., cast-iron radiator was used and curves from the data indicate that if the total heat emitted by a clean unpainted radiator is called 100, a coat of aluminum paint will cause the total heat discharged to drop to 88, and a coat of black paint will increase it to 103.

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[1] Professor of Mechanical Engineering, University of Michigan, Ann Arbor, Mich.
[2] Assistant Professor of Mechanical Engineering, University of Michigan, Ann Arbor, Mich.
[3] *Proceedings*, Third Annual Convention, *National District Heating Association*, 1911, p. 51, Coefficiencies of Heat Transmission.
[4] TRANS. A.S.H. & V.E., 1920, Vol. 26, p. 19. Heat Losses from Direct Radiation, J. R. Allen.
[5] TRANS. A.S.H.&V.E., 1920, Vol. 26, p. 27, Radiant Heat Given Off by a Direct Radiator, Allen and Rowley.

Presented at the Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago Ill., January, 1929.

About two years ago, additional data were presented[6] to this society by Prof. Severns which support the contention that flake metal paints reduce the heat emission. Quantitative values were given applying to a three-column, 6-section, Peerless radiator, 32 in. high, with 27 sq ft of surface. From these data the following figures showing relative performance are assembled for comparison with the data previously publis' ed and with those about to be presented:

Finish	Relative Heat Emission
Bare iron, foundry finish	100.0
One coat aluminum bronze	90.4
Gray paint dipped	100.6
One coat dull black Pecora paint	99.6

Radiators which are used for house heating are essentially convectors of heat.

FIG. 1. ARRANGEMENT OF APPARATUS

The heat carried away by air currents rising over and between the sections of the radiator is a larger percentage of the total heat dissipated than is the amount radiated. But the relative amounts of heat given out by convection and by radiation are dependent upon the form of the radiator. One made of a single horizontal pipe would discharge a much larger fraction of the total heat by radiation than would a wide, high radiator of many columns.

Within the last few years some important changes have been made in radiator design and, since the older style was used in the experimental work previously reported, it is believed that additional data obtained with one of the newer designs will be of interest.

General Plan

Two identical radiators are installed as shown in Fig. 1 in such a way that they can be operated in unison under the same conditions. One radiator is left

[6]TRANS. A.S.H.&V.E., Vol. 33, p. 41, Comparative Tests of Radiator Finishes, W. H. Severns.

unpainted to serve as a standard of comparison and the other is coated with the paint being investigated.

Each radiator has its own electrically heated boiler and is supplied with saturated steam at constant pressure and therefore constant temperature.

The heat input to each boiler is measured electrically and this heat is all dissipated by the apparatus. Subtracting the boiler and pipe line loss from the heat input gives the amount emitted by the radiator.

Test Room

The room used for the experimental work has certain characteristics of symmetry as shown in Fig. 2. It is 24 ft 4 in. by 22 ft 0 in. and 12 ft 8 in. high. Only one wall is exposed to outdoor weather conditions and this has two large windows 13 ft 4 in. apart, center to center. These windows and all openings into the room are kept closed during all tests. The room has no furniture except a table and a couple of chairs.

Description of the Radiators

The two radiators are alike, each being a standard 10-section, 26-in. four-tube, cast-iron, capitol radiator, having a nominal heating surface of 27.50 sq ft and an actual surface of 27.12 sq ft. The total length of the 10 sections is 25 in. and each section is 6¾ in. wide.

Locations of the Radiators

One radiator is installed under each of the two windows in the test room. Instead of resting directly on the floor of the room, each radiator is placed on an 8 ft by 6 ft platform 6 in. above the floor to give sufficient elevation for the return lines.

A false wall of beaver board extending up as high as the window sill, behind the radiator, with a 3½ in. air space between it and the brick wall of the building, provides a smooth, unbroken surface from the platform to the window sill, and protects the radiator to some extent from the effects of varying brick-wall temperatures. The radiator stands 2½ in. from the false wall with its top 7 in. below the window sill.

Boilers and Pipe Lines

The construction of the boiler is shown in Fig. 3. It is made of pipe and pipe fittings and is enclosed in a wooden box tightly packed with magnesia pipe covering to reduce heat loss as much as practicable.

A Westinghouse heater of the bayonet type is located near the bottom of the boiler in such a position that it is completely immersed in the water.

The steam passes from the upper section of the boiler to the top connection of the radiator and the water returns from the bottom connection, at the opposite end of the radiator, to the lower section of the boiler. Thus a complete circuit is provided and each radiator with its boiler constitutes a small, compact heating plant.

The pipe lines are well insulated with hair felt.

Temperature

Thermometers are suspended in the room at the locations shown in Fig. 2,

with the bulbs 5 ft above the floor. There are eight such thermometers and the average of the eight is taken as the "room temperature."

At locations 3, 4 and 8, thermometers are also placed one foot above the floor.

Thermometers are also located so as to give the temperature of the air entering the radiator near the bottom, the temperature of the air six inches above the radiator, and the outdoor temperature.

Heat Supply

Saturated steam is furnished to each radiator by its boiler. The temperature of the steam supplied is maintained constant by means of a sensitive automatic

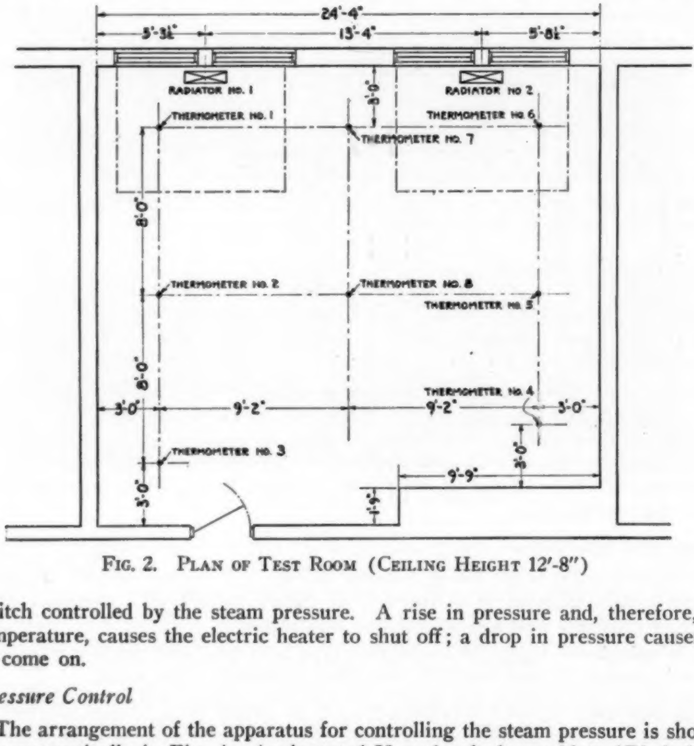

FIG. 2. PLAN OF TEST ROOM (CEILING HEIGHT 12'-8")

switch controlled by the steam pressure. A rise in pressure and, therefore, in temperature, causes the electric heater to shut off; a drop in pressure causes it to come on.

Pressure Control

The arrangement of the apparatus for controlling the steam pressure is shown diagrammatically in Fig. 4. An inverted Y made of glass tubing (*D*) is held in position by an adjustable clamp. The upper leg of the Y is connected by rubber tubing (*A*) to the steam line from the boiler. Each of the other two legs is connected by rubber tubing (*E*) to a mercury bottle (*E*). The liquid head of mercury and the pressure in the steam line are always in balance. An electric conductor (*G*) is immersed in the mercury (*F*) and when the two legs of mercury join in (*D*) a 6-volt electric circuit is completed through the relay

(*J*). When the mercury divides in (*D*) this circuit is broken; with the low voltage, there is very little arcing in (*D*).

The relay (*J*) is used to close or open the 220-volt circuit through the contactor switch (*K*). When the switch (*K*) is closed the heater in the boiler is *on* and when the switch is opened the heater is *off*.

Suppose that the switch (*B*) is closed and the equipment in use; the pressure of the steam is as shown by the manometer (*S*); the mercury in the two legs of (*D*) is not bridged over but just fails to join. The heater is *off*. A slight drop in steam pressure is accompanied by a rise in the level of the mercury in (*D*). When the two legs join at (*D*) the heater comes *on* and the steam pressure rises and soon shuts the heater *off* again by breaking the circuit at (*D*).

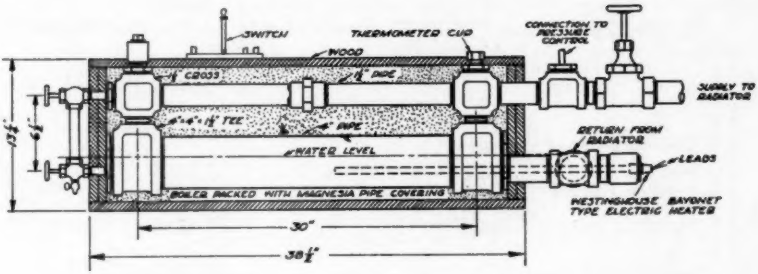

FIG. 3. ELECTRICALLY HEATED BOILER

Thus heat is applied intermittently to maintain constant pressure and therefore constant temperature steam.

Heat Measurement

All the electrical energy supplied to the heater is dissipated as heat by the radiator, pipe lines and exterior surfaces of the boiler. This energy is measured by a special recording portable induction test meter, manufactured by the General Electric Co., and designated as Type IB-6y. The recorded revolutions of the meter can be converted into British thermal units. Two such meters are used, one for each boiler.

After obtaining the gross heat dissipated, the valves at the radiator connections are closed, the radiator is vented to atmosphere and the heat loss of the pipe lines and external surfaces of the boiler is determined by supplying heat electrically as before. This heat, designated as boiler and pipe line loss, is subtracted from the gross heat dissipated, to give the net heat emitted by the radiator alone.

Paints

Five different paints have been used in the experiments, *viz.:*

(*A*) *Aluminum Paint*, manufactured by The Sherwin-Williams Co., Cleveland, Ohio. This paint consists of metallic pigment and a vehicle called Bronzing Liquid which is essentially a hard drying varnish thinned down to secure a proper working consistency. It is advertised as a protective, heat and water-resisting coating for metal

surfaces as gas engines, radiators, hot-water heaters, steam pipes, etc. It gives a bright silver finish.

(*B*) *Flat Brown Paint,* "Mellotone," manufactured by Lowe Brothers Co, Dayton, Ohio. This paint is known as Brown 618. It is designed to produce a flat finish and has the following composition:

Lithopone	30.5%
Silica and Silicates	20.8%
Lead chromate	8.3%
Ferric oxide	2.4%
Calcium sulphate	4.9%
Calcium carbonate	0.7%
Carbon	0.2%
Linseed oil	10.7%
Thinning japan	21.5%
	100.0%

(*C*) *Flat Tone—Cream.* This paint, manufactured by the Sherwin-Williams Co. consists of:

Pigment, by weight	69%
Liquid	31%
	100%

The composition of the pigment is:

Lithopone	82%	Zinc Sulphide	28%
Magnesium Silicate	18%	Barium Sulphate	72%
	100%		100%

Composition of liquid:

Linseed Oil	29%
Drier	14%
Mineral Spirits	57%
	100%

(*G*) *Liquid Gold,* manufactured by The Sherwin-Williams Co. The description given above for Aluminum Paint applies also to Liquid Gold. It gives a bright gold finish.

(*W*) *Vitralite,* manufactured by Robert Ingham Clark & Co., London, and described as White-Gloss Enamel.

A surface painted with aluminum paint when magnified one hundred times has a rough, irregular appearance, somewhat like the surface of a leveled pile of anthracite coal when view from a distance of about 50 ft.

A coating of liquid gold magnified one hundred times has the same characteristics of roughness as the aluminum paint, but on account of its color it resembles a closely packed mass of brass shavings.

Flat tone cream paint presents a surface under the microscope which is much smoother than those of the flake metal paints. It has the appearance of cake frosting. Flat brown paint gives a similar surface but a little rougher like the top of ginger-bread.

These surfaces have the same appearance whether the paint is sprayed on or is applied with a brush.

Tests

Test No. 1 was a preliminary test and is therefore not reported.

In Test No. 2 both radiators were unpainted and the performance reported is for clean cast-iron, foundry finish. For Test No. 3 one of the radiators was given a single coat of aluminum paint sprayed on. Preceding the next test a coat of gold bronze was applied over the aluminum. Two tests (Nos. 4 and 5) were made with this finish and then a coat of flat tone cream was added for Test No. 6. Then aluminum was applied on top of the cream for Tests 7, 8 and 9. Test 10 was made with an added coat of brown and Test 11 with white enamel on the brown. Having built up six coats, all of different paints and all sprayed on, it was decided to remove all of the paint and start again on some check tests.

After a thorough cleaning, down to the bare metal, two coats of white enamel were applied with a brush for Test No. 12. Next, a coat of aluminum, followed

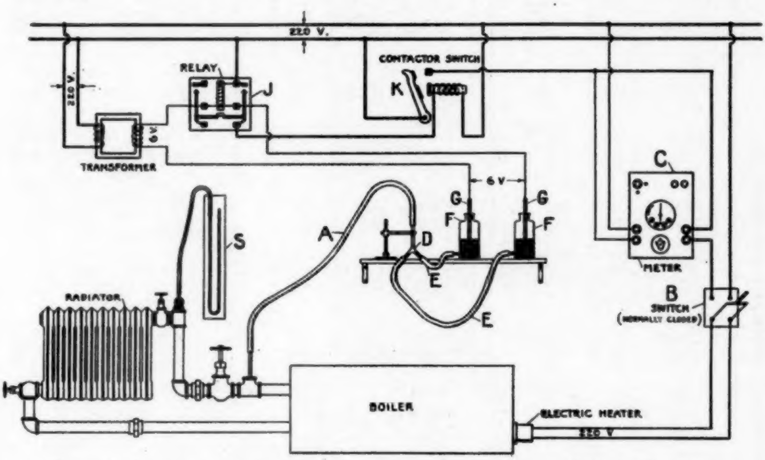

Fig. 4. Diagram of Apparatus for Controlling the Steam Pressure

by brown, then gold and finally cream, each sprayed on, was applied for Tests, 13, 14, 15 and 16, respectively.

Heat was kept on the apparatus for several hours preceding any test in order to establish conditions of equilibrium. The actual test was in no case less than one hour duration; usually it was longer. Simultaneous temperatures at the eight 5 ft. elevation seldom differed by as much as 1 deg. Readings were taken at 10-minute intervals, and the test was continued until the meter readings recording the heat input showed consistent results for at least an hour. For further assurance the tests were frequently extended over several hours. The instruments used were calibrated.

Results

Table 1 is a summary of the results. All the tests made, except Test No. 1, are reported. The radiator left unpainted to serve as the standard throughout the tests is designated as No. 1, and the other one which has been given various paint finishes is called No. 2.

TABLE 1. SUMMARY OF RESULTS

(a)	Test number	2		3		4		5		6	
(b)	Radiator number	1	2	1	2	1	2	1	2	1	2
(c)	Steam temperature F	215.0	215.0	215.0	215.0	214.9	215.0	215.0	215.0	215.0	215.0
(d)	Room Temperature F	80.5	80.5	85.6	85.6	82.3	82.3	81.9	81.9	75.8	75.8
(e)	Temperature difference F (c—d)	134.5	134.5	129.4	129.4	132.6	132.7	133.1	133.1	139.2	139.2
(f)	Paint	....	None	....	A	....	A-G	....	A-G	....	A-G-C
(g)	Btu per hour, gross	7137	7100	6855	6293	7025	6404	7053	6549	7842	7831
(h)	Boiler and piping loss, Btu per hour (actual)	802.2	817.0	844.1	809.2	792.1	824.9	884.9	830.0	1223.4	926.8
(i)	Temp diff. during boiler and piping loss test	138.7	138.7	131.6	131.6	134.8	135.0	136.0	135.9	145.5	145.5
(j)	Boiler and piping loss per degree temp diff. $(h \div i)$	5.7837	5.8904	6.4141	6.1489	5.8761	6.1104	6.5067	6.1074	8.4083	6.3698
(k)	Boiler and piping loss adjusted to temp diff. prevailing during radiator test (j) x (e)	778	792	830	796	779	811	866	813	1170	887
(l)	Btu per hour emitted by radiator, net, $(g - k)$	6359	6308	6025	5497	6246	5593	6187	5736	6672	6944
(m)	Relative heat emission $\frac{\text{Radiator No. 2}}{\text{Radiator No. 1}}$		0.9124		0.9124		0.8948		0.9272		1.0407
(n)	Relative heat emission, Bare C. I. Foundry Finish taken as 100		100		92.0		90.2		93.5		104.9
(a)	Test number	7		8		9		10		11	
(b)	Radiator number	1	2	1	2	1	2	1	2	1	2
(c)	Steam temperature F	215.0	215.0	215.0	215.0	215.0	215.0	215.0	215.0	215.1	215.1
(d)	Room temperature F	81.5	81.5	81.6	81.6	79.1	79.1	84.6	84.6	84.0	84.0
(e)	Temperature difference F $(c - d)$	133.5	133.5	133.4	133.4	135.9	135.9	130.4	130.4	131.1	131.1
(f)	Paint	....	A-G-C-A	....	A-G-C-A	....	A-G-C-A	....	A-G-C-A-B	....	A-G-C-A-B-W
(g)	Btu per hour, gross	7203	6584	7290	6670	7299	6744	6708	6852	6929	7082
(h)	Boiler and piping loss, Btu per hour (actual)	1148	851.0	1140.5	878.0	992.4	880.1	877.9	792.1	828.1	809.1
(i)	Temp diff. during boiler and piping loss test	136.7	136.7	139.1	137.4	138.9	138.9	133.4	133.4	133.4	133.5
(j)	Boiler and piping loss per degree temp diff. $(h \div i)$	8.3983	6.2252	8.1991	6.3898	7.1447	6.3362	6.5810	5.9378	6.2077	6.0607
(k)	Boiler and piping loss adjusted to temp diff. prevailing during radiator test (j) x (e)	1121	831	1094	852	971	861	858	774	813	795
(l)	Btu per hour emitted by radiator, net $(g - k)$	6082	5753	6196	5818	6328	5883	5850	6078	6116	6287
(m)	Relative heat emission $\frac{\text{Radiator No. 2}}{\text{Radiator No. 1}}$		0.9459		0.9390		0.9296		1.0389		1.0281
(n)	Relative heat emission, Bare C. I. Foundry Finish taken as 100		95.3		94.7		93.7		104.7		103.6
(a)	Test number	12		13		14		15		16	
(b)	Radiator number	1	2	1	2	1	2	1	2	1	2
(c)	Steam temperature F	215.0	215.0	215.0	215.0	215.0	215.1	215.0	215.0	215.0	215.0
(d)	Room temperature F	81.4	81.4	80.2	80.2	79.7	79.7	82.2	82.2	78.7	78.7
(e)	Temperature difference F $(c - d)$	133.6	133.6	134.8	134.8	135.3	135.4	132.8	132.8	136.3	136.3
(f)	Paint	....	W-W	....	W-W-A	....	W-W-A-B	....	W-W-A-B-G	....	W-W-A-B-G-C
(g)	Btu per hour, gross	7149	7151	7041	6590	7154	7251	6993	6626	7346	7503
(h)	Boiler and piping loss, Btu per hour (actual)	752.8	754.2	746.9	790.3	976.7	812.4	775.9	816.3	814.1	820.8
(i)	Temp diff. during boiler and piping loss test	136.3	136.3	136.4	136.4	138.6	138.7	135.5	135.5	138.8	138.8
(j)	Boiler and piping loss per degree temp diff. $(h \div i)$	5.5231	5.5334	5.4758	5.7940	7.0469	5.8573	5.7262	6.0244	5.8653	5.9136
(k)	Boiler and piping loss adjusted to temp diff. prevailing during radiator test (j) x (e)	738	739	738	781	953	793	760	800	799	896
(l)	Btu per hour emitted by radiator, net $(g - k)$			6303	5809	6201	6458	6233	5826	6547	6697
(m)	Relative heat emission $\frac{\text{Radiator No. 2}}{\text{Radiator No. 1}}$	6411	6412 1.00		0.9217		1.0405		0.9346		1.8229
(n)	Relative heat emission, Bare C. I. Foundry Finish taken as 100		100.8		92.9		104.9		94.2		103.1

The steam temperature is determined from the pressure shown by the combined readings of the barometer, and the manometer on the steam line supplying the radiator. The room temperature is the average of the readings of the eight thermometers located as previously described.

For convenience the kind of paint is designated by its initial letter, such as *A* for aluminum, *B* for brown, *C* for cream, *G* for gold and *W* for white. The designation *A-G-C-A* means a coat of aluminum followed by a coat of

TABLE 2. CONDENSED SUMMARY

Test No.	Paint on Radiator No. 2	Heat Emission Relative
2	None	100.0
3	*A*	92.0
7	*A-G-C-A*	95.3
8	*A-G-C-A*	94.7
9	*A-G-C-A*	93.7
13	*W-W-A*	92.9
		Av. 93.7
4	*A-G*	90.2
5	*A-G*	93.5
15	*W-W-A-B-G*	94.2
		Av. 92.6
6	*A-G-C*	104.9
16	*W-W-A-B-G-C*	103.1
		Av. 104.0
10	*A-G-C-A-B*	104.7
14	*W-W-A-B*	104.9
		Av. 104.8
11	*A-G-C-A-B-W*	103.6
12	*W-W*	100.8
		Av. 102.2

gold, then cream, then aluminum again, making a composite of four successive coats in the order given.

The heat emitted is reported for the temperatures prevailing during the test and is not corrected to a basis of 70 F room temperature.

If the conversion factors adopted by the A.S.H.V.E. are applied to both radiators their relative performance will not be changed.

However, the curves of Fig. 5 which show the heat dissipated per hour plotted against room temperatures are extended to 70 F.

Only the final coat influences the heat emission, as evidenced by Tests 5 to 11 inclusive. In Test 5 the finishing coat on Radiator No. 2 was liquid gold and the relative performance 93.5. For Test 6 a coat of cream was added and the relative performance increased to 104.9. On top of the cream a coat of aluminum was applied for Tests 7, 8 and 9, and the relative performance dropped again to 94.6 (average of 95.3, 94.7 and 93.7). When a coat of brown was added for Test 10 the relative performance increased again to 104.7. With white

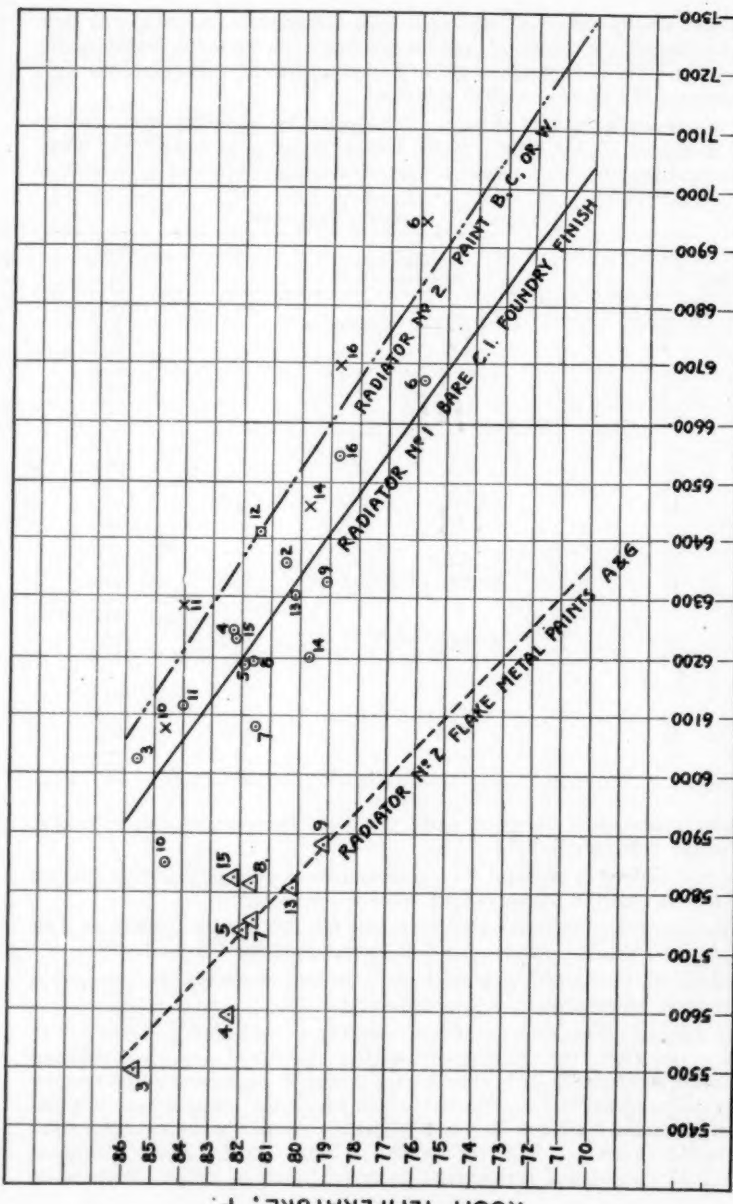

FIG. 5. SHOWS THE HEAT DISSIPATED PER HOUR PLOTTED AGAINST ROOM TEMPERATURES

applied on top of the brown for Test 11 the relative performance was 103.6, practically unchanged.

Table 2 is arranged to group the data according to the final finishing coat. The only results that show much disagreement with the others are those for Tests 4 and 12. If Test 4 is omitted and the liquid-gold group included with the aluminum group, the average for all tests with flake metal paints becomes 93.8; if Test 4 is included the average is 93.3. Similarly, if the last three groups are combined, the average, omitting Test 12, is 104.2; if Test 12 is included the average is 103.7.

Fig. 5 shows the heat dissipated per hour plotted against room temperature. The lines converge somewhat for increased room temperatures. Presumably they should curve slightly and join when the temperature of the air surrounding the radiator equals the temperature of the steam, and the heat transfer is zero.

Most of the tests were conducted when the room temperature was in the neighborhood of 80 to 82 F. At 80 F if radiators having the proportions of the ones used in these tests are given a surface coat of flake metal paint, their ability to emit heat will be about 10 per cent less than if the surface coat is one of the colored pigment paints described in this paper. At a room temperature of 70 F, taking the values from Fig. 5, the reduction would be about 13 per cent.

DISCUSSION

M. T. CLOW (WRITTEN): Would there be any differences between the effect of surface paints on the older style of radiation and on the newer style, if the ratio of radiating (or enveloping) surface to actual surface is the same, and if so, why? Since the effect of a surface paint on radiator performance is dependent upon the proportions of the radiator, it seems to me that it would be more convenient to express the effect of the surface paint on the heat transmission of the radiating instead of the actual surface, which effect would be the same for every radiator. Then, to determine the effect of any surface paint on the heat transmission of the actual surface of any radiator, multiply the increase or decrease in the heat transmission of the radiating surface by the ratio of radiating to actual surface.

I would like to ask the authors to explain the variations which occur in the results obtained from several tests of the same surface paint. For instance, why is it that Test No. 7 shows a smaller reduction in heat transmission of aluminum paint than test No. 3 in spite of the fact that the room temperature was lower in Test No. 7 than in Test No. 3? Also, why is the boiler and piping loss higher for radiator No. 1, Test No. 3 than for radiator No. 1, Test No. 2, in spite of the fact that the temperature difference between the steam and the room is lower in Test No. 3?

What is the variation in steam temperature between the opening and closing of the circuit in the electrical control of the apparatus?

Would one be justified in drawing the following general conclusion from the test data reported in this paper? All flake metal paints reduce the heat transmission of cast-iron radiators and all other paints increase the heat transmission of cast-iron radiators.

On page 121 of the paper under the heading *Temperature*, it is said that the

temperature of the air was taken as it entered the radiator at the bottom and as it left the radiator at the top. I would like to ask the authors what these temperatures were. I think such temperatures are imporant because they may be indicative of the relative heating effect of high and low, wide and narrow radiators.

H. E. LONGWELL (WRITTEN): While I am not convinced that the paper proves very conclusively anything about the effect of surface paints on radiator performance, I, nevertheless, regard it as a most interesting and edifying contribution, in that it demonstrates so forcibly the difficulties and uncertainties of measuring with any considerable degree of accuracy, the heat emission of the conventional cast-iron radiator.

At first sight the arrangements for testing appear to be ideal. Yet on analysis, the results of the experiments show variations that are decidedly violent, erratic, and difficult to account for.

Confining our attention for the moment to the series of fifteen tests on radiator No. 1, which is unpainted, we should expect the hourly gross heat loss per degree of difference between 215 F and the temperature of the room, to be fairly constant. Yet in Test No. 10 the gross heat loss per deg is 51.442 Btu while in Test No. 6 it is 56.336 Btu—a difference of 9½ per cent. The difference between the steam temperature and the room temperature in Test No. 6 is substantially 9 deg higher than in Test No. 10. Naturally we would expect some increase in the heat loss per degree, with the increasing temperature difference, because of the more rapid circulation of the air, but we have believed that this difference does not exceed 2 per cent for an increase of 10 deg in the temperature difference between the steam and the surrounding air. The comparison between Tests No. 10 and No. 6 opens up the interesting speculation as to whether the effect of the difference between room temperature and steam temperature may not be several times as great as we have suspected.

Similarly we should expect the boiler and piping loss per degree of temperature difference to be fairly uniform, provided of course, that the *set-up* was not changed during the entire series of tests, and the same boiler was used in the same location throughout, as good practice demands in investigations of this nature. However, this loss per deg appears to be 5.4758 Btu in Test No. 13, and 8.3983 Btu in Test No. 7, an increase of over 53 per cent, with temperature differences varying less than one-half degree.

In six of the tests the heat loss from boiler No. 1 is less than from boiler No. 2, and in the 9 remaining tests the heat loss from boiler No. 2 is the lower. This circumstance in connection with the fact that the boilers are apparently identical in every detail, would lead us to expect that there should be no significant difference in the heat loss between the averages of each of the two series of fifteen tests, each pair having been made simultaneously and under the same conditions. Nevertheless the average of the series made on the No. 1 boiler, is substantially 10 per cent greater than the average of the series made on the No. 2 boiler.

In Test No. 5 the greater heat emission of the painted radiator is accounted for entirely by the apparently greater heat loss from the No. 1 boiler. As a matter of fact the reported gross heat output of the No. 2 radiator and boiler combined, is even slightly less than that of the No. 1 unit.

It must be admitted that the computed results of these tests show remarkable agreement with our accepted theories about the effect of radiator paints; but it is difficult to account for this agreement on any hypothesis other than that of accidental coincidence. Test results that vary as much as 10 per cent, are not very convincing for measuring differences which lie wholly between the limits of 0 and 8 per cent.

It might be helpful to study again the test set-up in order to see if we can discover any probable weaknesses that might account for the apparently illogical results of the experiments. The following points are suggested for consideration:

1. Since the test for the heat loss from the boiler is made by isolating the radiator by means of stop valves on the supply and return pipes, a trifling leak in either or both of these valves might have a marked effect on the apparent measurement.

2. The heat loss from the boiler, as reported is, in terms of electrical units, a very small quantity, ranging between 0.22 and 0.36 kw per hour. Is the electric meter used, sufficiently delicate and sensitive to measure such small quantities within a relatively negligible limit of error? Even the maximum load of boiler and radiator combined does not exceed 2.2 kw per hour.

3. The supply of electric current is intermittent, and the duration of these periods of supply, and the length of the intervals between them are not stated. Is it certain that the meter will sum up these intermittent supplies of energy with the same degree of accuracy that it would register the total of a continuous flow throughout the test period?

4. The radiators under test are located immediately beneath windows. Might it not be possible that changing weather conditions, especially the direction and force of winds might alter the actual heat emission to a measureable degree? Is it not possible that there might be a marked and variable difference in the infiltration in and about these windows?

Personally I should be inclined to suspect first the accuracy of the electric meter as a measure of energy delivered at frequent intervals in periods of perhaps unusually short duration. This may be a condition not usually met with in meter practice, and it may not have been fully investigated by meter engineers. I should suspect that the inertia of the meter mechanism might seriously affect the integrity of the measurements.

In conclusion, I fear—very much to my regret—that it would be hopeless for me to ask the authors to believe that I have not reviewed this paper in any hostile spirit, but such is really the case. This Society has become so accustomed to receiving reports of *results* of tests, that it is refreshing to encounter a report, which, regardless of possible consequences, is not stripped of its originally observed data, but affords anyone who is interested, an opportunity to make his own computations and deductions, with a full knowledge of the amount of *selection* and *rationalization* that is necessary to line these data up into smooth orderly curves. I believe that progress would be more rapid, and discussion more active, interesting and constructive, if we had more reports telling the *whole* story as this one does; and I believe that the Society is richer by reason of its presentation.

From my own experience, I doubt if we really have any data on the heat emission of radiators that is much more consistent than that contained in this paper.

The method of test appealed to me most strongly as offering the best prospects for obtaining data that would satisfy the most captious critic. Perhaps I

am unduly disappointed over the apparent results so far. Nevertheless I am still convinced that the authors are on the right track, and that eventually some further development or refinement of this plan will give us more dependable, complete and satisfying knowledge in this line of research, than we have ever had before.

A. I. BROWN (WRITTEN): The authors of this interesting paper state that its purpose is two-fold; namely, to describe the methods and to present the results.

The method is of particular interest, and in the matter of manipulation of apparatus for repeated tests, it appears less cumbersome than the common method of weighing condensation. A study of the data obtained in the fifteen tests of the bare radiator No. 1, however, suggests the question as to whether the results are as accurate as those obtained by the more common and more direct method.

The total amount of heat supplied to the boiler, piping, and bare radiator No. 1, per degree difference between the steam and room temperature ranges from 51.4 to 56.3 Btu. Variation in these values is due partly to variation in the temperature differential, as indicated by a comparison of tests Nos. 6 and 3. The temperature differential in test No. 6 is 9.8 F, or 7½ per cent higher than in test No. 3, but the total heat loss tor test No. 6 is more than 14 per cent higher than in test No. 3. Comparison of tests Nos. 14 and 3, however, show practically equal rates of heat loss per degree, namely, 52.8 Btu for test No. 14 and 52.9 Btu for test No. 3, although the temperature difference between the steam and room is 5.9 F or 4.55 per cent higher for test No. 14 than for test No. 3.

These differences in rate of heat loss may be due to differences in humidity of the atmosphere for which no data are recorded in the paper, but this factor cannot account for the great differences in the values shown by the loss from boiler No. 1 and piping. The value of 1070 Btu for test No. 6 is apparently a typographical error and should be 1170 Btu, and this value is 58½ per cent higher than the value of 738 Btu for tests Nos. 12 and 13. This variation amounts to more than 6 per cent of the average gross amount of heat supplied to the boiler piping and radiator in the fifteen tests. It is significant to note that a sudden rise in the values of heat loss from boiler and piping occurs in test No. 6 and the values then gradually decrease to a minimum in test No. 13. One might suspect a small leak in the radiator valves, which of course could be detected by a drain connection in the radiator, but such a leak would hardly seem probable with steam at such a low pressure, and it is possible that the variation in values is due to some changes in piping or heat insulation which the authors have not mentioned.

Corresponding values of heat loss from Boiler No. 2 and piping are much more uniform, but still the variation between maximum and minimum values amounts to more than 2 per cent of the average gross heat supplied to the boiler, piping, and radiator.

These unaccounted-for variations of possibly 6 per cent or of even 2 per cent are larger than should be expected in tests of the effect of paint on radiator performance.

The residual heat of an electric heater of the type used is probably of the order of 50 Btu, and if the temperature of the heating element is not the same

at the end as at the beginning of the test, an error of this magnitude may result. Two tests with the error in opposite directions would show an appreciable disagreement. It would, therefore, seem to be essential to start and end all tests at a definite interval of time after the electrical circuit is closed or opened. Otherwise, this chance for error could be reduced by rheostat regulation of the electric input to the heater so as to reduce to a minimum the amount of residual heat in the heating element.

The authors' illustrations of the arrangement of apparatus, Figs. 1 and 4, show the manometers and pressure regulating switch located above the radiators. The connecting tubes, several feet in length, may be filled either with steam or with condensation in uncertain amounts, thereby influencing the indication of pressure. A similar riser, connection, 10 ft in height and made of pipe as large as ¼ in. standard, is known to have entirely filled with condensation. As a precaution against this source of error a preferred location for such manometers is below the connection into the steam line, so that a definite column of water exists on one side of the manometer.

In radiator testing by the more common method of weighing the condensate, the greatest source of error is usually that due to variations in the quality of steam supplied. Here, however, the amount of moisture in the steam supply seldom is more than 5 per cent, and a good separator in the line will easily remove 90 per cent of this moisture; the remaining moisture, when assumed to be dry steam, therefore represents less than ½ of one per cent of the total heat supplied. Where high-pressure steam is available even greater accuracy is possible by separating the moisture from the high-pressure line and then throttling so as to maintain a few degrees of superheat near the radiator inlet valve. Even 10 F of superheat represents an increase in heat content of less than ½ of one per cent. With no element of uncertainty, but by a little adjustment of the pressure at the steam separator or by variation in the amount of heat insulation on the steam supply line, it is not difficult to regulate the degree of superheat to as low as 2 or 3 F.

The method described in this paper, however, surely warrants further investigation and appears to involve no inherent difficulties which would prevent a high degree of accuracy. It is hoped that further development and refinement of this scheme will make it possible to duplicate results with greater accuracy and facility than are possible with present methods.

F. B. ROWLEY: I think the subject has been very well covered and is clear in most of our minds.

Some years ago we performed some similar experiments at the University of Minnesota in which we placed the radiators in a vacuum in order to separate the radiant from the convected heat. We found that practically all of the difference that is shown for the various paints was due to the reduction in the radiant heat. The amount of heat given off by convection was substantially the same in all cases. Our findings in the total reduction were similar to those given by Professor Fessenden.

H. P. REID: I would like to ask the authors whether or not these radiators were completely painted or painted only against the eye. In other words, the average radiator as painted in the home is only painted on the exposed surfaces, as shown. Were these radiators completely covered with paint? If so, would not this affect the application of these data?

E. K. CAMPBELL: In view of the fact brought out by Professor Rowley that this affects only the radiant heat of the radiator and not the convected, I would like to ask what is the net reduction in the heating value of the radiator as a whole.

E. B. ROYER: I would like to ask if anyone has figured the actual effective heat in this case. It seems to me that the whole story is not told by the condensation, as in the first paper this morning it was shown that it was not the condensation of a radiator that affected the comfort of the room.

It appears that aluminum paint not only decreases the condensation of the radiator but, further, makes an even greater difference than shown by the figures, in the actual comfort of the room, that is, you do not get as much radiant heat which is very effective.

M. BARRY WATSON: In this connection, I would like to inquire whether any attempt had been made to determine the difference due to surface paint with the low temperature radiators. This would apply in the case of hot water heating or in the case of low temperature steam heating systems which are now becoming more or less popular. It is generally conceded that the convected heat is nearly in proportion to the difference in temperature between the air and the radiator, whereas the radiant heat is not in that same proportion. Is this feature of surface paint an important feature in low temperature radiators, or is it of negligible importance?

C. H. FESSENDEN: Some fraction of the total heat emitted by the radiator is given off as radiant heat; the magnitude of the fraction is influenced by the form of the radiator. A low, long radiator would transmit by radiation a larger fraction of the total heat emitted than a high, short radiator having the same amount of surface.

Under the influence of the pressure control, the maximum variation in the manometer reading was about two-tenths of an inch, *i. e.*, plus or minus one-tenth. This corresponds to about a third of a degree in temperature.

The temperatures above and below the radiators were as follows:

Test No.	Temperatures, Degrees Fahrenheit			
	Radiator No. 1		Radiator No. 2	
	Above	Below	Above	Below
2	120.7	96.4	124.2	95.3
3	124.1	100.7	126.8	92.6
4	122.2	97.2	124.8	92.0
5	121.6	97.5	125.0	90.0
6	118.8	89.0	120.3	87.7
7	119.5	96.8	123.0	87.6
8	120.0	95.0	121.4	89.7
9	121.5	94.6	121.4	87.1
10	128.0	99.9	126.0	95.4
11	127.3	94.9	129.3	91.3
12	122.7	90.6	128.1	87.1
13	121.4	84.6	125.3	83.1
14	119.7	89.3	125.0	88.5
15	120.7	91.5	127.5	86.3
16	118.5	87.8	121.9	89.0

The entire surface of the radiator was painted.

I would not draw the general conclusion from these tests that *all* flake metal paints would produce the same effect as the two used in these experiments, nor that all non-flake metal paints would give the same results as the ones we used.

The boiler and piping loss for the unpainted unit (No.1) was not so uniform throughout the series of tests as the loss for unit No. 2. This was due in part to steam climbing a portion of the hose connection to the pressure regulator and, by keeping that part at the hose hot, adding to the heat loss. When this occurred it was present during the main test as well as during the boiler and piping loss test. Whether it was the same for both the main test and the piping loss test was a matter of judgment arrived at by noting the length of hose that was hot. In test No. 7 particularly, and to some extent in No. 8, the boiler and piping loss for the unpainted radiator seems to be high. This was taken into account in drawing the curves of Fig. 5.

MR. CAMPBELL: Were you referring to net reduction in the heating value of the radiator?

PROFESSOR FESSENDEN: No separation has been made between the radiant heat and the convected heat. The heat measured was the total heat emitted in any manner.

MR. CAMPBELL: Measured by condensation?

PROFESSOR FESSENDEN: No, we measured the heat supplied electrically and converted it to Btu.

The meters used were practically new. They were calibrated before and after the tests. One was correct within 0.1 per cent and the other within 0.9 per cent. The meter correction was included in computing the results.

The interval of operation was approximately every half minute; you will readily see, therefore, that no large amount of heat could remain stored in the heating element.

In testing the meters the intermittent operation was taken into account. The circuit was interrupted every half minute for a considerable period and the accumulated meter reading compared with the reading of the meter when operated continuously for the same total time; the difference was less than 0.2 per cent.

The two boilers are similar but not exact duplicates and the amount of piping connecting the boiler with the radiator is not exactly the same for the two installations. So it is improper to compare No. 1 boiler and piping loss with No. 2 boiler and piping loss.

The boiler and piping loss for the painted radiator shows less variation than for the unpainted radiator. While preparations were being made for one of the tests the hair-felt surrounding the heater connections on No. 1 piping caught fire and had to be renewed. Preceding test No. 12 the pipe lagging on both installations was improved.

The paper presented by Professor Kratz showed that the walls of the special test room absorbed heat over a long period and that about 24 hours was required to establish stable conditions. When a radiator is installed in an ordinary room where the temperature gradient through the walls cannot be controlled it will not necessarily emit the same amount of heat on two different days even

though the steam and air temperatures are the same during the two tests. In one case the walls may absorb more heat than in the other and the radiator must supply the extra demand. A twenty-four preliminary run with the expectation of establishing stable conditions would lead to disappointment because, with varying outdoor weather influences, a stable wall condition cannot be obtained. But if two radiators are tested at the same time they can be operated under the influences of the same conditions and the results are then comparable.

PRESIDENT WILLARD: It is a matter of very great satisfaction to find that Professors Fessenden and Marin, of the University of Michigan, are carrying forward the work so ably started many years ago by Prof. John R. Allen, Dean of the University, first director of the Research Laboratory of the Society, a distinguished investigator himself, a man who left his imprint, left his personality on this Society in a way that almost no other man has done. There has hardly been a session of the A. S. H. V. E. at which the University of Michigan has not had one or more conspicuous papers, contributions to our knowledge of the subject of heating and ventilation.

I wish to thank Professors Fessenden and Marin for their paper.

No. 823

ARCHITECTURAL ASPECTS OF CONCEALED HEATERS

By J. H. Milliken,[1] Chicago, Ill., *and* H. C. Murphy,[2] Louisville, Ky.

MEMBERS

THE development of the extended surface type of concealed heaters has shown remarkable progress in the last few years. No attempt will be made in this paper to consider details of design, construction or efficiency of the various types of heaters now on the market, nor is it the purpose of this paper to consider the relative efficiency of this recent development as compared with cast iron radiation. Numerous exhaustive and carefully conducted tests have shown this nonferrous type of extended surface heating unit to be entirely practical, and the consensus of opinion of architects, engineers and owners who have installed these new types seems to be favorable.

Rather, it is the intention of this paper to point out the possibilities which this new method of heating offers the engineer in meeting the growing demands from the architect and owner for a heating system that, while efficient, shall be unobtrusive, and which will not detract from the general decorative scheme.

There are certain fundamental differences between extended surface or convection heaters and exposed cast-iron radiators and certain claims of advantage attributed to the former must be considered by the designer in relation to its installation. It will be well at this point to give a typical descripion of this equipment and its operation. As far as the boiler and piping arrangements are concerned, there is no real difference. The same boiler size and essentially the same piping arrangement is used with either system. In fact it is quite usual to install the extended surface type of heater in some selected rooms of the house and cast-iron radiators in others, such as the kitchen and service rooms.

Broadly speaking, the new system is designed to heat by *convection* as opposed to heating by *radiation*. It is true that much of the heat emanated by the cast-iron radiator is transmitted by *convection*. In fact Allen and Rowley report in the Journal of the American Society of Heating & Ventilating Engineers, Vol. 26, p. 103, 1920, as high as 73 per cent by convection as against 27 per cent by radiation. The new heaters therefore are designed specifically—and with the principal purpose of utilizing to the fullest extent the convection

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currents of air as a vehicle in the distribution of heat. A diagram showing some typical heating units now on the market is given in Fig. 1.

These do not by any means represent all the different types available, but will assist in studying the fundamental principles involved.

The depth of the heating arrangement is fixed by standard building practice in that heaters are designed to go in walls having 2 x 4 in., 2 x 6 in. or 2 x 8 in. studding. While the thickness of the wall fixed the depth of the heater and consequently imposed that limiting dimension on the extended surface, the most efficient height of the fins or plates had to be determined by test and experiment. The spacing of the fins or plates was determined by the resistance to air flow and thickness of the plates. It was found, however, that the thickness of the fins or plates themselves had very little effect on the efficiency of heat transfer.

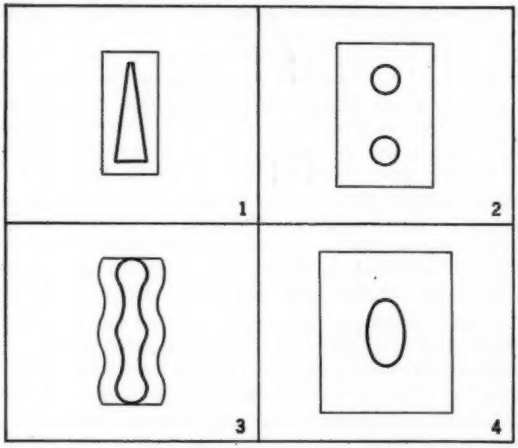

FIG. 1. DIAGRAM SHOWING TYPICAL HEATING UNITS

The heating element is made of a metal having a high coefficient of conductivity and carries on its outer surface extension plates or fins which rapidly effect the transfer of heat from the steam to the air between these extended surfaces. As the air is heated it rises rapidly, being displaced by the colder air which flows in the intake at the floor level. The heated air rises through a sheet metal duct or stack to the outlet grille, where a curved deflector projects it into the zone of occupancy. The diagram given herewith illustrates the method of distribution of heat by convection from the extended surface heater—or convector—as opposed to heat transfer by convection with the cast-iron radiator.

Investigation has shown that with the new method heat may be distributed much more evenly throughout the zone of occupancy often with a temperature difference of less than 5 F between floor and ceiling as compared with as much as 15 F with the cast-iron radiator depending, of course, upon the proper placement of heaters and outlet grilles.

The advantages claimed for this method are that it provides air circulation

in the room uniformly heating the zone of occupancy with elimination of excessive ceiling temperatures and overheating in the immediate vicinity of the heater.

A thorough investigation and comparison of the performance of cast-iron radiators with and without enclosures has been conducted by Professor Willard in the laboratories at the University of Illinois.

The chart given here indicates the comparative demands upon the boiler for heating up of a typical *convection* heating unit as compared with a cast-iron radiator. The radiator curve shown by a dotted line is taken from page 74 of the A.S.H.&V.E. GUIDE, 1928. The curve for the convection heater shown by the full line was developed by tests in the laboratory of a manufacturer of extended surface heaters. In considering boiler load THE GUIDE states:

It is often very important to know the maximum condensation that occurs in a radiator when steam is turned on. The chart shows the condensation rate in pounds per hour

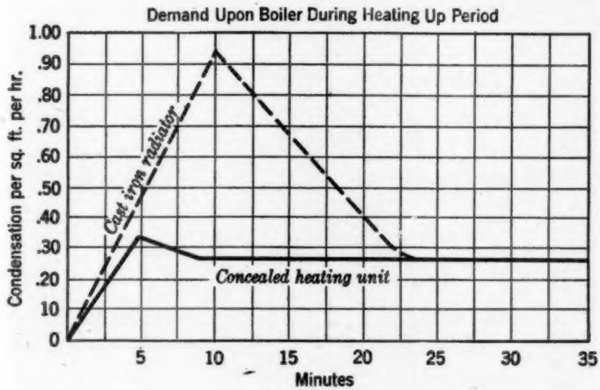

FIG. 2. DIAGRAM SHOWING CONVECTION HEATING

for the time elapsing after the steam is turned into the radiator. It will be noticed that the maximum condensation in the radiator occurs 10 min after steam is turned on, and in that case it amounts to about three and one-half times normal condensation. After the end of 25 min the radiator had reached a normal rate of condensation. This curve was made from observations at intervals of 10 min so that the intermediate points between the 10-min points are not known, and the form of the curve is not exact. It shows, however, that when a plant is initially started up and the system filled with steam during a short period of time the demand made upon the boiler may be very much higher than the normal demand. In practice the rate of steam supply to the radiator while heating up is, however, frequently retarded by controlled elimination of air through air valves or traps and if sufficient time is allowed for the heating-up process, overload on the boiler may be practically eliminated.

It will be noticed that the maximum condensation with the extended surface heater occurred five minutes from the time steam was turned on and that the normal rate of condensation was reached in eight minutes.

In heating up the convection heater unit there is, of course, much less air to be displaced than in the cast-iron radiator and a consequent shortening of the time required for heating up—five minutes as compared to 25 min—and for the

same reason the convection heater requires less time to cool when the unit is shut off.

Many ingenious systems of piping, valves, traps, etc., have been developed to insure proper heat regulation and modulation. Most of these systems were handicapped in their operation by the cast-iron radiator because of the great mass of iron which had to be heated—and cooled—before the regulation became effective, frequently causing impatient comments about the "janitor," the "boiler" or the "Heating system," and an attempt to regulate the heat by opening the window. With the extended surface heater the response to control is more rapid, and positive heat regulation and modulation is readily accomplished. There is not the usual waiting for the "radiator" to heat up—or cool off—something that is especially desirable in bedrooms where the heat is shut off at night but required immediately upon arising.

With the brief description given of the basic differences and essential requirements of this new heating method, the details of the installation and the method

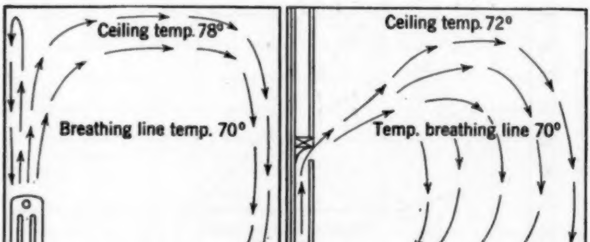

FIG. 3. CURVE SHOWING DEMANDS ON BOILER

of applying it to concrete problems can be considered. While the design of various units now available naturally differs in many respects, the writers have selected a type with which they are familiar, and which is fairly representative of this new type of equipment.

Stack effect is of the utmost importance in this new development. The intensity of its action depends upon the height of the stack and the difference in temperature between the heated air and the air entering the units. The efficiency of the units therefore depends considerably upon proper installation and consideration in the layout and design.

In general the length of the heater and the height of the stack are dependent upon the available wall space and the heating capacity required within reasonable limits, the higher the stacks the greater the capacity of the heater, provided the stacks are well insulated from heat losses.

In consideration of the foregoing the writers believe that for this method to be successful it is essential that there be close cooperation between the building designer and the heating manufacturer so that proper locations and proper allotments of space can be made for the necessary heaters. For instance, if the plans are completed and concealed heaters are then considered, it may be found that

some spaces available for the heater are so narrow as to require a stack height in order to have the necessary Btu delivery that will be inconsistent with the best circulation in the room or *vice versa.* Furthermore if the installation of the heaters is considered in the initial design the outlet grilles can be worked up in a harmonious arrangement with other openings and decorations in the rooms to render them entirely inconspicuous. Also, wall details may be many times arranged so that increased wall thickness can be obtained where heaters

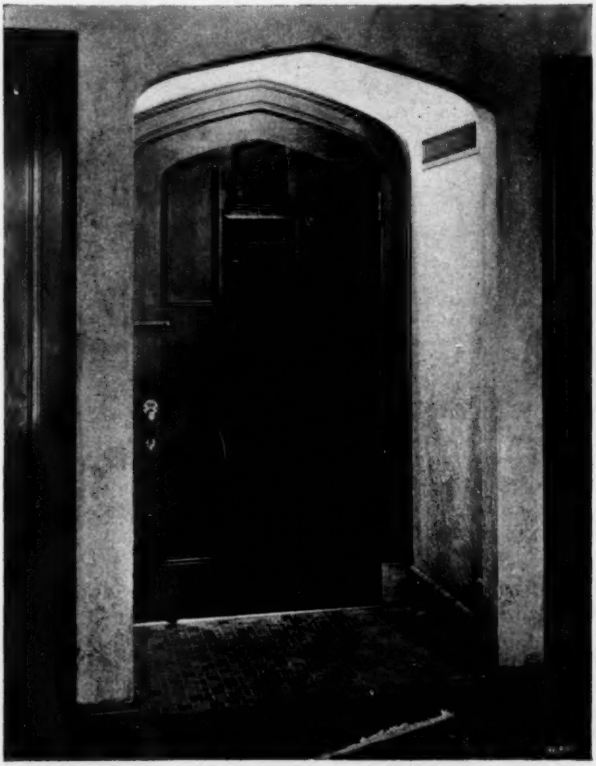

FIG. 4. ENTRANCE HALL EQUIPPED WITH CONCEALED HEATERS

of large size are required thereby cutting down the spread of inlet and outlet openings.

Heating the entrance hall has always presented a problem. Heat is needed—plentifully—but sufficient space for large heating units is frequently unavailable. The photograph, Fig. 4, illustrates the method used by an engineer in the middle west. The hall floor is of colored tile; the inlet is framed in tile; and the outlet grille harmonizes well with the decorative scheme. The heating unit is entirely concealed in the wall, does not occupy any floor space and has given

FIG. 5. RECESS FRAMED IN WALL TO ACCOMMODATE HEATER AND STACK

entire satisfaction both to the architect and owner. Note the well proportioned inlet and outlet openings, a feature which should be given careful attention. Sometimes in an effort to secure an especially attractive outlet or inlet grille pattern, a design is used which materially cuts down the free opening at the inlet or outlet. This seriously reduces the efficiency of the heater. The recommendations of the heater manufacturer should be followed very carefully in this respect. The installation of concealed heaters involves very little additional planning or labor, but as the procedure of framing in, placing the heater and the piping, lathing and plastering are new to some workmen, strict attention to the printed instructions accompanying the heater should be observed. The photographs, Figs. 5 to 7, show the progressive steps of framing in, placing of the heater and piping and plastering in.

The heating installation is finished well in advance of other parts of the build-

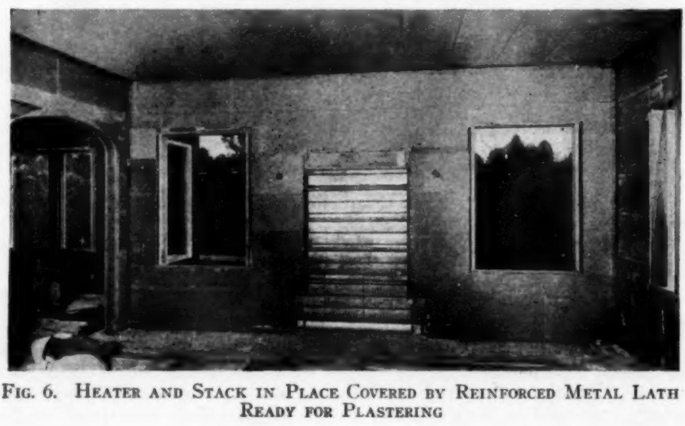

FIG. 6. HEATER AND STACK IN PLACE COVERED BY REINFORCED METAL LATH READY FOR PLASTERING

ing work—before the plastering, interior woodwork, etc., allowing heat to be supplied when necessary in the progress of building by the heaters in their permanent position.

There are many situations where "in the wall" heaters solve a difficult situation—for instance the installations shown by Figs. 8 and 9 offer a practical solution of a difficult problem.

In this connection it might be well to consider the conveniences possible with the substitution of convection heaters in tall cabinets where existing radiator installations are unsatisfactory due to their radiant effect upon the surrounding

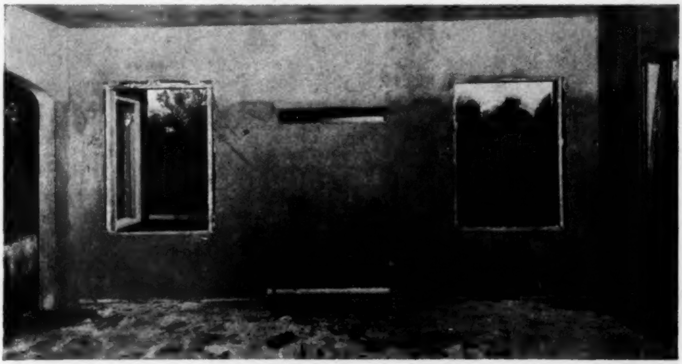

FIG. 7. UNIT PLASTERED IN WITH INLET AND OUTLET READY FOR GRILLES

furniture or occupants of the room. It is often possible at a relatively small expense to prevent overheating in the immediate neighborhood of exposed radiators and eliminate poor circulation of heat throughout the room. The convection heater makes not only the space near the heater more comfortable, but assists materially in the proper circulation of heat through the space of occupancy. Even when a radiator under a window is found objectionable the adjacent spaces can be made more comfortable and healthful by extending the steam pipes to the nearest available wall space for the installation of a heater in a cabinet.

Fig. 10 illustrates the convenience and beauty of a concealed heater in a bathroom, its saving in space—where space is generally at a premium.

Many owners of fine apartment buildings find that their apartments rent quicker with concealed heaters. The treatment of an apartment lobby in New York City is shown in Fig. 11. The black enameled inlet grille blends perfectly with the black marble baseboard; the outlet grille, enameled white, is unobtrusive. The breakfast room shown here has a removable wall panel for access to the piping connections.

The concealed heater has wide appeal to designers and interior decorators as they have an ever-increasing tendency to harmonize the necessary mechanical utilities of the home with the other architectural treatment.

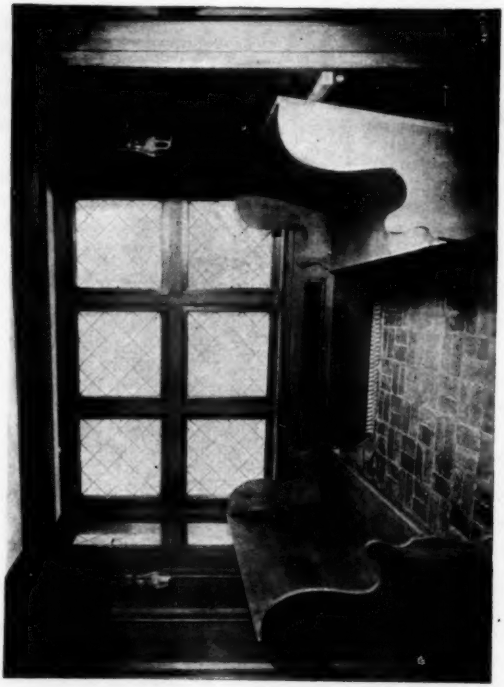

FIG. 9. INSTALLATION OF CONCEALED HEATER IN NOOK

<—FIG. 8. INSTALLATION OF CONCEALED HEATER IN STAIRWAY

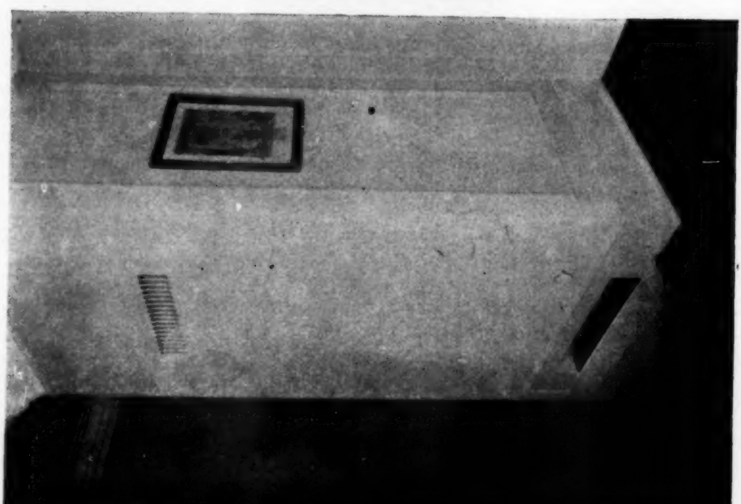

FIG. 10. BATHROOM INSTALLATION OF CONCEALED HEATERS

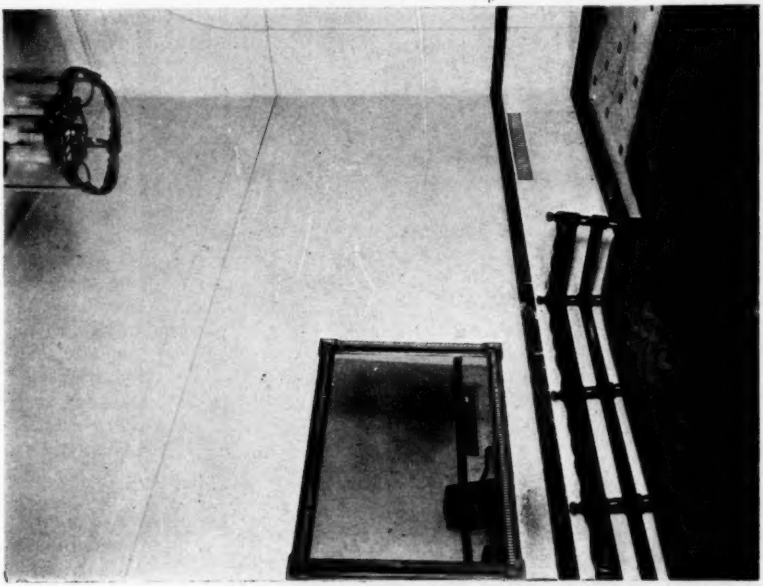

FIG. 11. CONCEALED HEATER IN APARTMENT LOBBY

FIG. 12. INSTALLATION OF CONCEALED HEATERS WITH REMOVABLE ACCESS PANELS IN BREAKFAST ROOM

DISCUSSION

PROF. J. D. HOFFMAN: May I inquire if it is a desirable thing to have concealed heaters within an outside wall. The last slide gave a dining room with heaters enclosed. Is that desirable or undesirable?

J. H. MILLIKEN: There would be no objection at all to putting heaters in the outside wall providing the outside of the stack for carrying the heat is insulated. The heat loss through the stack and outside wall would render the heater ineffective. To obtain proper circulation of heated air, you would have to take into consideration the location of stairways, openings, and the height of ceilings. It is possible, where, for instance, a heater is placed in a narrow space beside some opening that a very high stack might be put in and bring the air out very high in the room, and, considering the size of the room, might throw the point of delivery too high in the room, and perhaps not get the benefits of the circulation. You would then have a high ceiling temperature and insufficient heat below the breathing line.

W. H. CARRIER: There are several features connected with concealed radiation that I think heating engineers should take into consideration.

The performance of concealed radiation of the extended surface type is quite different in many respects from that obtained by the standard cast-iron radiation whether unenclosed or concealed. In the first place, there is not only a very much lower peak load in the heating-up period, as was pointed out in the paper by Messrs. Milliken and Murphy, but also there is a very much more rapid rate of heating up during this period due to smaller steam space and to much lower mass of material. Roughly, the time of heating up is about one-tenth that of cast-iron radiation. However, it must be realized that there is a correspondingly quicker period of cooling off. It is this striking difference in the performance of the extended surface copper radiation that gives it its great advantage in some respects, while in other respects it may be a disadvantage unless the peculiarities of its performances are taken into consideration in the lay-out of the system. It is on this account that one of the essential elements to be taken into consideration in the proper design of concealed radiation of the extended surface type is that the steam distribution is such that the radiators more remote from the boilers will heat up just as quickly as the radiators near the boilers. This is especially necessary in large residences having extensive horizontal distributing mains. One method of accomplishing this is to carry the main steam line nearly to the end of the system and then bring the steam back. Particular attention must also be given to the quick venting of the air from the main distributing lines so that steam will be supplied immediately to all radiation. If these precautions are not taken, unsatisfactory operation may result in residences with automatic temperature control owing to the fact that certain portions of the building will be overheated while other portions will be underheated due to the intermittent operation of the control.

The effect of different types of radiators, I think, has been pretty well discussed here, and it points in two rather diverse directions or extremes, both away from the present old style radiation; one, as Dr. Brabbée very ably points out, the purely radiant type, where you get the maximum per cent of radiation possible, installed under windows, radiating to the room, giving a very high economy, and apparently very uniform heat distribution.

A reverse method of arriving at the same result was pointed out in the paper presented by Professor Kratz. In this case radiator shields were used which, which reducing the total amount of radiation effect, actually increased the economy of heating and gave better distribution of temperature owing to a systematic increase of air circulation over the surface.

Concealed radiation operates entirely on the latter principle utilizing no radiation effect whatever. It is not certain as yet which method gives the greater increase in economy over the old type of radiation. This can only be determined by laboratory tests. The prime requisite, however, of all heating systems is human comfort and the two distinct types of heating, one primarily by radiation and the other entirely by controlled convection, each have their advantages and disadvantages.

When a person comes in out of the cold, he likes at first to get near some source of radiant heat, such as a fire-place or radiant type of heater, where he can be warmed up rapidly. This radiant heat effect, however, soon passes from one of comfort to one of discomfort and a person is obliged to move away from

the source of radiant heat. Long and close proximity to any source of radiant heat causes discomfort, and it is certain that a greater proportion of the room is made habitable by the convection method of heating as contrasted with the radiant method of heating. These, however, are two diametrically opposed methods of heating, both tending toward economy and more uniform distribution of temperature, and it appears as though we were going to abandon the old type of radiator for either one or the other of these newer types. Both of these types also greatly improve the appearance of the room and the æsthetic side of heating is nowadays being emphasized.

I believe in heating our homes, the first thing we consider is comfort; the second, appearance, and the third, economy.

J. C. MILES: This paper seems to cover what I used to term the unit heater. I had quite a bit of money at one time, and spent it on a unit heater. In looking at some of the slides, it looked to me as though this is not a radiator; we are not discussing radiation, but a convection heater, or a convector.

My mind runs along the lines of convection heat as the healthful form of heat according to nature, and I think the idea here is right in so far as it goes, but when you enclose a radiator, you add a lot of friction, and you reduce your air volume or current over the radiator. I notice the arrows coming out showing how they operate, and I am reminded of an engineer who used to show arrows all over the house, and one of his designers made one and opened the kitchen transom and ran the arrows out to the driveway and heated the smokehouse that way.

The question of reduction of ceiling temperature and raising of the floor temperature is entirely dependent up on the air turnover. I think that if there is any investigation made on this, you will find that the larger volume of air and the lower difference between the incoming air temperature and the floor temperature, the more uniform will be your temperature from floor to ceiling, so when we restrict the circulation over-radiation, like we do over-enclosed radiators, we will perhaps have to put some mechanical circulation back of it. I wonder if Mr. Carrier's troubles would not be overcome if we could keep steam inside this coil and control our condensation by the speed of the air over the coil.

H. M. HART: I would like to emphasize the importance of the proper locations of these radiators, from experience in my own office. I constructed one of these flue type radiators, out of a cast-iron radiator, by putting a shield around it, with the outlet about 4 ft 6 in. above the floor. The radiator was located on the inside wall at right angles to two large windows. I thought I had a fine arrangement, because I was going to cut off the radiant heat which hit me in the back and made me uncomfortable. That is why I enclosed it. The result was that I had cold feet. The cold air from the windows would drop to the floor and seemed to create a great difference in temperature between the breathing line and the floor line. So I took the radiator out and put a long low radiator in front of the windows; no more radiation, but I obtained comfort in that way, indicating that consideration should be given to the location of the unit. I have had no experience with this type of installation but I have questioned the advisability of some types that were illustrated where the inlet was near the ceiling on an inside wall. Placing of the unit under the window, I think, would be preferable.

MR. MILLIKEN: Undoubtedly, if the outlet from one of these heaters is

placed near the ceiling, you are going to lose a great deal of benefit of the control of air circulation. You are getting about the same effect as you do with the exposed cast-iron radiator, and you do not have the direct radiation to warm your feet, as you might in the case of the cast-iron radiator. So that an arrangement such as the one with the outlet at the ceiling would produce very poor circulation.

H. C. MURPHY: The point made by Mr. Miles relative to damper control is a good one. A number of units of this type are available in which the flow of air through the unit is controlled by a modulating damper placed either directly over the heating element or at the outlet grille.

It is possible for instance to leave the heat on with the modulating damper closed in certain selected rooms; for instance, bathroom, dressing room, etc., and secure immediate delivery of heat upon rising in the morning by simply opening the damper. With the damper closed the heat losses can be held to approximately 10 per cent of the normal output of the unit. This is quite a usual application and is in fact installed in the bathroom installation illustrated in this paper, but was not specifically described.

P. F. CALLAGHAN, JR.: I noticed that all the installations shown were more or less permanent in the, you might say, architectural aspects; that is to say, the radiator was plastered into the wall. Is that considered an ideal installation?

Every one knows a vacuum system or vapor system requires that the traps be cleaned, I believe, semi-annually. The valves may need attention in the course of several years, and that would mean ripping out the wall in this type of installation.

There are, of course, other types of installations and the question I wanted to raise was whether the permanent type was the best.

MR. MILLIKEN: There are arrangements on the market for giving access to the heater so that you can do anything with the traps or piping connections that you want or in fact, remove the heater and replace it if necessary, and, of course, it is essential that in order to have this access, either a metal plate be installed on the wall or else a frame work be provided so that a plaster plate can be installed and taken off. A great many installations are made without any such access. In that case, of course, the traps are placed below the floor, and the heaters are used both ways. I believe it would be hard to say just which would be the last word at this time.

No. 824

ADDITIONAL COEFFICIENTS OF HEAT TRANSFER AS MEASURED UNDER NATURAL WEATHER CONDITIONS

By F. C. HOUGHTEN,[1] (*Member*) CARL GUTBERLET[2] (*Non-Member*),
AND C. G. F. ZOBEL[3] (*Member*)

PITTSBURGH, PA.

HEAT flow through walls under natural weather conditions has been studied by the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATIG ENGINEERS for several years, resulting in a number of reports in the Journal. This paper gives the results of further study on a number of types of construction.

The Nicholls' Heat Flow Meter of the Research Laboratory was used for measuring the rate of heat flow through the walls. A complete description of the meter, the method of applying it to a wall, the observations made, and the method of analyzing result appears in other laboratory reports[4] and will not be repeated here.

Definitions, Nomenclature and Symbols for Heat Transmission

The engineer interested in heat transfer through building material has used a number of terms or coefficients having definite physical significance. The names of such coefficients and the symbols used to represent them, have not, however, been uniform and definite. Different engineering groups have used their own nomenclature and symbols, and while a number of attempts have been made at standardization, little has been accomplished.

The Heat Transmission Committee of the National Research Council has recently adopted a standard of nomenclature and symbols which will, it is hoped, be adopted by all engineers interested in heat flow, whether through building materials, through boiler tubes or electrical insulators.

It seems in keeping with progress toward standardization that the Research

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[3] C. G. F. Zobel, Research Physicist, Research Laboratory, A.S.H.&V.E.
[4] P. Nicholls, Measuring Heat Transmission in Builing Structures and a Heat Transmission Meter, TRANS. A.S.H.&V.E., Vol. 30, 1924, pp. 65-104. F. C. Houghten and C. G. F. Zobel, Coefficients of Heat Transfer as Measured Under Natural Weather Conditions, TRANS. A.S.H.&V.E., Vol. 34, 1928. F. C. Houghten and C. G. F. Zobel, Heat Transfer Through Roofs Under Summer Conditions, TRANS. A.S.H.&V.E., Vol. 34, 1928.

Presented at the Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, Ill., January, 1929.

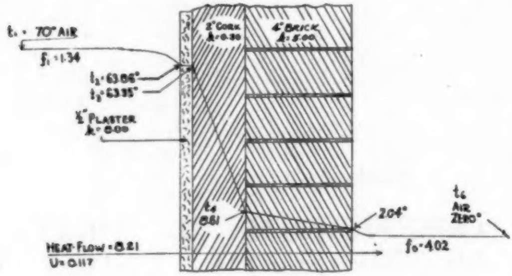


FIG. 1. TEMPERATURE GRADIENT THROUGH INSULATED BRICK WALL

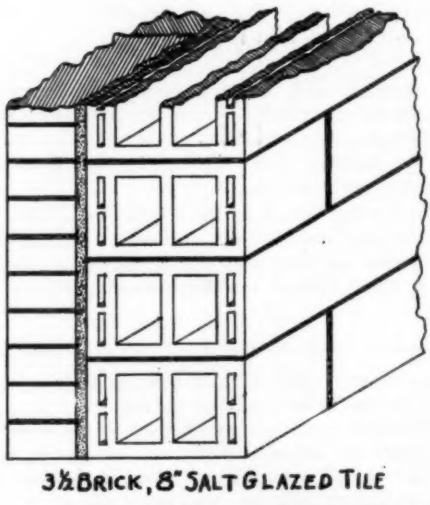
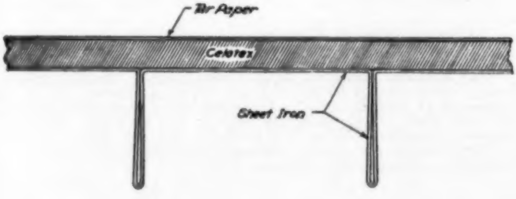


FIG. 2. TYPE OF BRICK AND HOLLOW-TILE WALL TESTED

FIG. 3. STEEL DECKED ROOF



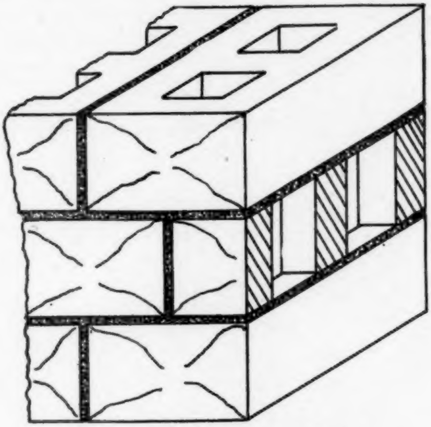


Fig. 4. A 12-In. Concrete Block

Laboratory should adopt the recommendation of the National Research Council, and this is done in this report with the exception of the symbol for conductance, which it seems desirable for the sake of simplicity to change. An attempt will be made to have the Research Council adopt this change.

The coefficients in which Heating and Ventilating engineers are interested

Brick Veneer Wall

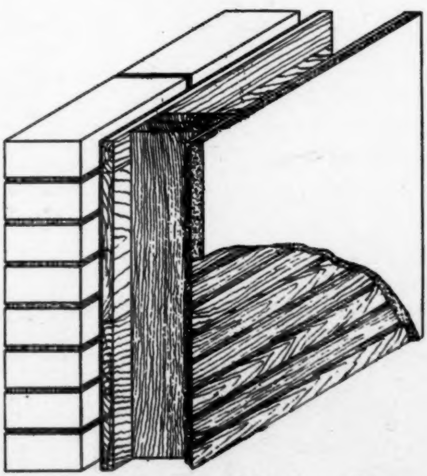


Fig. 5. A 4-In. Brick, 1-In. Space, Paper, ⅞-In. Sheathing, Studs, Lath, Plaster

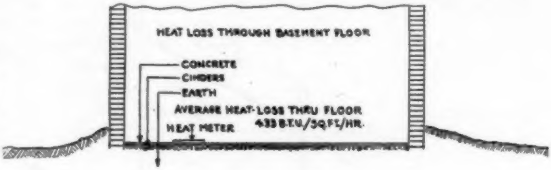


FIG. 6. SECTION OF BASEMENT FLOOR THROUGH WHICH HEAT FLOW WAS MEASURED

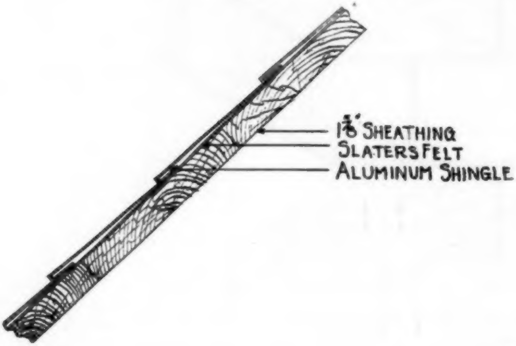


FIG. 7. ALUMINUM SHINGLE ROOF

SCHENLEY APTS.

Date	Exposure	C_1	ΔC	C	Exposure	C_1	ΔC	C
Jan 11	North	–	–	–	East	0.38	−0.04	0.34
Jan 12		0.34	−0.03	0.31		.39	−.05	.34
Jan 13		.37	−.06	.31		.44	−.10	.34
Jan 14		.22	+.08	.30		.29	+.05	.34
Jan 15		.24	+.05	.29		.27	+.07	.34
Jan 16		.31	−.02	.29		.35	−.04	.31
Jan 17		.38	−.06	.32		.43	−.09	.34
Jan 18		.35	−.05	.30		.40	−.07	.33
Jan 19		.25	+.05	.30		.39	−.02	.37
		Average	0.30			Average	0.34	

C_1 = Partial Conductance
ΔC = Correction for Heat Capacity
C = Conductance B.T.U. per Sq. Ft. per Hour per °F.

FIG. 8. 8-IN. TILE, BRICK VENEER WALL

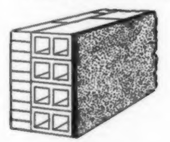


FIG. 9. PRIVATE RESIDENCE WITH CONCRETE BLOCK AND BRICK VENEER WALLS

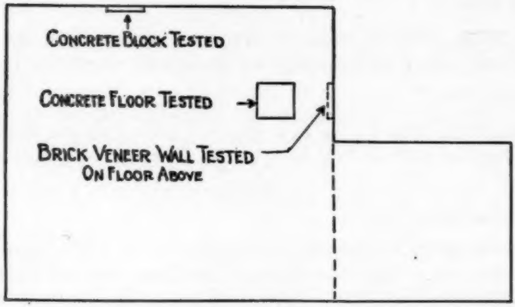


FIG. 10. LOCATION OF WALLS TESTED IN SMALL RESIDENCE

are: conductivity of homogeneous materials, conductance of entire walls or parts of walls, total transmission through a wall from air to air, and surface coefficients. Besides these coefficients, the heating engineer is interested in other terms and symbols.

Temperature

The engineer is interested in temperatures at various points along the path of heat flow through a wall, and has used either t or T, for this purpose. The recommendation of the National Research Council for temperature in either degree fahrenheit or degree centigrade is t. Several temperatures along a path are expressed at t, t_1, t_2, etc. T is used to express temperature in deg Absolute.

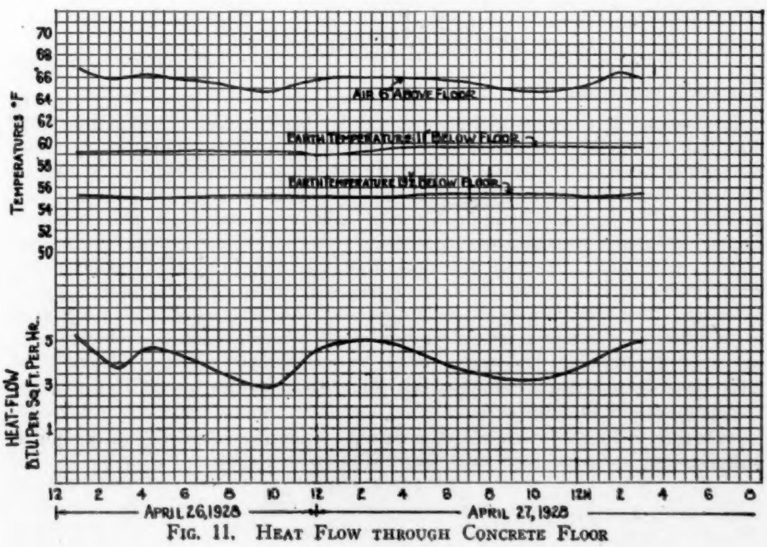


FIG. 11. HEAT FLOW THROUGH CONCRETE FLOOR

Rate of Heat Flow

Instead of using either H or h, the new standard uses q as the rate of heat flow along a path and Q as the quantity transferred over a given time.

Length of Path

In the place of S, D or L, the new standard uses L as the distance between two points along the path of heat flow, as the thickness of a wall or component part thereof.

Thermal Conductivity

Thermal conductivity is defined as the property of a homogeneous material expressed as the rate of heat flow through unit area and unit thickness in unit time with unit temperature drop. In place of the various symbols K, k, C and c. used heretofore, the new standard is k.

Referring to Fig. 1 it will be noted that:

$$\text{Thermal Conductivity of brick} = k_B = \frac{qL_B}{A(t_4 - t_5)} = \frac{(8.211 \times 4)}{1(8.61 - 2.042)} = 5.0$$

$$\text{Thermal Conductivity of cork} = k_C = \frac{qL_C}{A(t_3 - t_4)} = \frac{(8.211 \times 2)}{1(63.35 - 8.61)} = 0.3$$

$$\text{Thermal Conductivity of plaster} = k_P = \frac{qL_P}{A(t_2 - t_3)} = \frac{(8.211 \times 0.5)}{1(63.86 - 63.35)} = 8.0$$

Thermal Conductance

Conductance is a very useful term to the heating engineer in calculating heat flow through walls. In the English system of units it is defined as the heat transmitted through the entire wall or through a component part of a wall in Btu per square feet hour per degree temperature difference between the two surfaces. The symbol C has been used by many in the past. The Research Council in its new standard has adopted the reciprocal of the symbol for thermal resistance, or $1/R$. In many branches of engineering, little use is made of the term conductance, and the adoption of the more awkward reciprocal is not a hardship. Its greater use in the field of heating and ventilation, however, makes it very desirable to use a separate symbol, and it would seem desirable to prevail upon the National Research Council to amend their report to include it in their recommended standard. Pending such decision, the term C for thermal conductance will be continued.

Referring to Fig. 1 it will be noted that the thermal conductance of the entire wall is:

$$C = \frac{1}{R} = \frac{q}{A(t_2 - t_5)} = \frac{8.211}{1(63.86 - 2.042)} = 0.133$$

Film Conductance (Surface Transmission)

The terms surface coefficient or surface transmission coefficient and others have been used in the past to express the heat transfer between the surface of a wall and the air in contact with it, or *vice versa*, k, k_1, k_2, etc., have been used as symbols. The Research Council has adopted the term Film Conductance, represented by the symbols f, f_1, f_2, etc., for several surfaces, or f_i and f_o for inside and outside surfaces of a wall.

Referring to Fig. 1 it will be noted that:

$$\text{The inside film conductance } f_i = \frac{q}{A(t_1 - t_2)} = \frac{8.211}{1(70 - 63.86)} = 1.34$$

$$\text{The outside film conductance } f_o = \frac{q}{A(t_5 - t_6)} = \frac{8.211}{1(2.042 - 0)} = 4.02$$

Transmittance (Total Transmission)

The National Research Council has adopted the term Transmittance defined as the heat passing through the wall per square feet of area per hour per degree temperature difference between the air on the two sides of the wall. The universally used symbol U was adopted in the new standard.

Referring to Fig. 1 it will be noted that:

$$U = \frac{q}{A(t_1 - t_6)} = \frac{8.21}{1(70 - 0)} = 0.117$$

or

$$U = \frac{1}{\frac{1}{f_i} + \frac{L_P}{k_P} + \frac{L_C}{k_C} + \frac{L_B}{k_B} + \frac{1}{f_o}} = \frac{1}{\frac{1}{1.34} + \frac{1/2}{8} + \frac{2}{0.3} + \frac{4}{5} + \frac{1}{4.02}} = 0.117$$

Types of Construction Tested

1. Commercial Garage in Pittsburgh.
   a. 3½-in. brick, 8-in. salt glaze tile. (Fig. 2)
   b. Roof—steel decked, ½-in. Celotex, 5-ply Roofing Felt. (Fig. 3)
   c. ½-in. Celotex in above roof.
2. Ten-Room Private House in Suburb.
   a. Cellar wall—12-in. concrete block. (Fig. 4)
   b. Brick veneer wall. (Fig. 5)
   c. Basement concrete floor. (Fig. 6)
   d. Cinder fill in above floor.
   e. Earth under above floor.
3. Ten-Room Private House in Suburb.
   a. Aluminum shingle roof. (Fig. 7)

Test Results

Table 1 gives the description of and the coefficients for the different walls studied. The results compare favorably with those appearing in earlier Laboratory reports and values given in Chapter 1 of THE GUIDE, 1928. In several instances the type of construction differs and this must be taken into account in comparing the values given in this report with others.

The transmittance coefficient U for the brick-hollow-tile wall tested was found to be 0.16. A similar wall tested in the Schenley Apartments and previously

reported[5] by the Laboratory gave a coefficient of 0.21. This may be accounted for by the difference in construction of the wall and the tile used. The wall and tile here reported had the construction shown in Fig. 2 while the wall and tile in the brick-hollow-tile construction in the Schenley Apartments is shown in Fig. 8. The lower coefficient found in this report can be accounted for by the extra air spaces in the tile and the lack of plaster.

Taking the transmittance $U = 0.21$ for the construction shown in Fig. 8 and adding the thermal resistance of one additional air space and subtracting the resistance of the plaster we have the calculated value,

$$U = \frac{1}{\frac{1}{0.21} + \frac{1}{1.34} + \frac{1}{1.34} - \frac{0.5}{8.0}} = 0.162$$

The above computation which accounts for the difference in transmittance between the walls in Figs. 2 and 8 is based upon the assumption that the two rows of small air spaces on either side of the tile are equivalent to one continuous air space, (each row of small spaces actually covers $^7/_{11}$ of the wall area), having two film conductance coefficients of 1.34).

TABLE 1—HEAT TRANSMISSION DATA

PROJECT	DATE	DAYS	CONSTRUCTION
LIBERTY AVE. GARAGE	APRIL, 1928	5	3½" BRICK, 8" SALT GLAZED TILE.
		2	STEEL DECKED ROOF, ½" CELOTEX, TAR PAPER.
		2	CELOTEX - ½"
FIVE ROOM PRIVATE HOUSE	APRIL, 1928	8	12" CONCRETE BLOCK
		8	4" BRICK, PAPER, ⅜" SHEATHING, STUDS, LATH, PLASTER
		1	CONCRETE FLOOR
		1	CINDERS 2"
		1	EARTH 9½"
SEVEN ROOM PRIVATE HOUSE	AUGUST, 1928	11	ALUMINUM SHINGLE, PAPER, 1⅝" SHEATHING

C—Conductance—Btu per square foot per hour per degree fahrenheit difference between surface temperatures.
f_i—Inside film conductance—Btu per square foot per hour per degree fahrenheit difference between inside air and surface temperatures.
f_o—Outside film conductance—Btu per square foot per hour per degree fahrenheit difference between outside air and surface temperatures.

The transmittance coefficient, $U = 0.395$, reported for the insulated steel deck roof compares very favorably with the coefficient, $U = 0.358$ in THE GUIDE. This roof was insulated with ½ in. of Celotex, which was found to have a conductance of 0.65, giving a thermal conductivity of 0.325 compared with 0.33 given in THE GUIDE.

The concrete block wall, the brick veneer wall, and the basement floor studied were in the private residence shown in Fig. 9. Fig. 10 shows the location of the different constructions tested in this house.

The 12 in. hollow-concrete-block wall gave a transmittance of 0.29 which is in close agreement with THE GUIDE value of 0.287. However, the test here

[5] Coefficients of Heat Transfer as Measured Under Natural Weather Conditions, by F. C. Houghten and C. G. F. Zobel, TRANS. A.S.H.&V.E., Vol. 34, 1928.

reported was made with practically no wind blowing and consequently the outside film conductance, $f_o = 2.65$, is low compared with the value, $f_o = 4.02$ used in THE GUIDE computations. Also, the conductance of the block wall tested was higher than used in THE GUIDE. The value here reported being $C = 0.59$ giving an apparent conductivity for the entire block including air spaces of $k = 7.08$, compared with, $C = 0.403$, and $k = 4.84$ used in THE GUIDE computations. This higher value of k for concrete is in keeping with several test results recently reported,[6] all indicating that we are using too low values for concrete. This is further evidence that conductivity of concrete and variation in conductivity with mix, type of aggregate, etc., should be the subject of further investigation.

The brick veneer wall reported had a transmittance, $U = 0.17$ as compared with $U = 0.216$ for a similar wall reported in THE GUIDE. The wall here reported, Fig. 5, had a 1-in. air space between the sheathing and brick in place of the ½ in. cement mortar in the wall given in THE GUIDE.

DEVELOPED WITH NICHOLLS' HEAT METER

EXPOSURE	No. of Tests	C	f_i	f_o	U	k	t_m	IN CONDITIONS	OUT CONDITIONS
S.E. WALL	1	0.26	1.12	0.71	0.16	—	44.5	GLAZED TILE SURFACE	3.4 M.P.H. WIND
FLAT ROOF	2	0.635	1.69	2.81	0.39	—	45.6	RIBBED SURFACE	
INSULATION	2	0.65	—	—	—	0.325	—	BETWEEN STEEL, & PAPER	
S.E. WALL	1	0.59	0.90	2.65	0.29	7.08*	51.7	CONCRETE BLOCK SUR.	0.9 M.P.H WIND
S.W. WALL	1	0.22	1.61	1.25	0.17	—	61.9	NEW PLASTER SURFACE	0.9 M.P.H. WIND
	1	—	—	—	0.39	—	—	SMOOTH CONCRETE	STILL AIR
UNDER CONCRETE	1	1.16	—	—	—	2.32	—		
UNDER CINDERS	1	0.89	—	—	—	8.46	—		
SLOP. ROOF	2	.362	5.14	2.37	—	—	82.4	NEW PLANED SURFACE	1.5 M.P.H. WIND

U—Transmittance—Btu per square foot per hour per degree fahrenheit difference between inside air and temperatures.
*—Apparent conductivity of block including air space.
tm—Mean temperature degrees fahrenheit.
k—Conductivity—Btu per square foot per hour per degree temperature difference for 1-in. thickness.

THE GUIDE value of 0.216 may be corrected for the air space by replacing the ½ in. plaster by subtracting the thermal resistance of ½ in. plaster and adding an air space resistance made up of two film resistances having film conductances of 1.34.

$$U = \frac{1}{\frac{1}{0.216} - \frac{0.5}{8} + \frac{1}{1.34} + \frac{1}{1.34}} = 0.165$$

This correction, like the one on the hollow tile wall above, gives a very close agreement between the value found by test and the computed value in THE GUIDE and lends confidence to the application of the accepted method of comput-

[6] Coefficients of heat transfer measured under natural weather conditions.

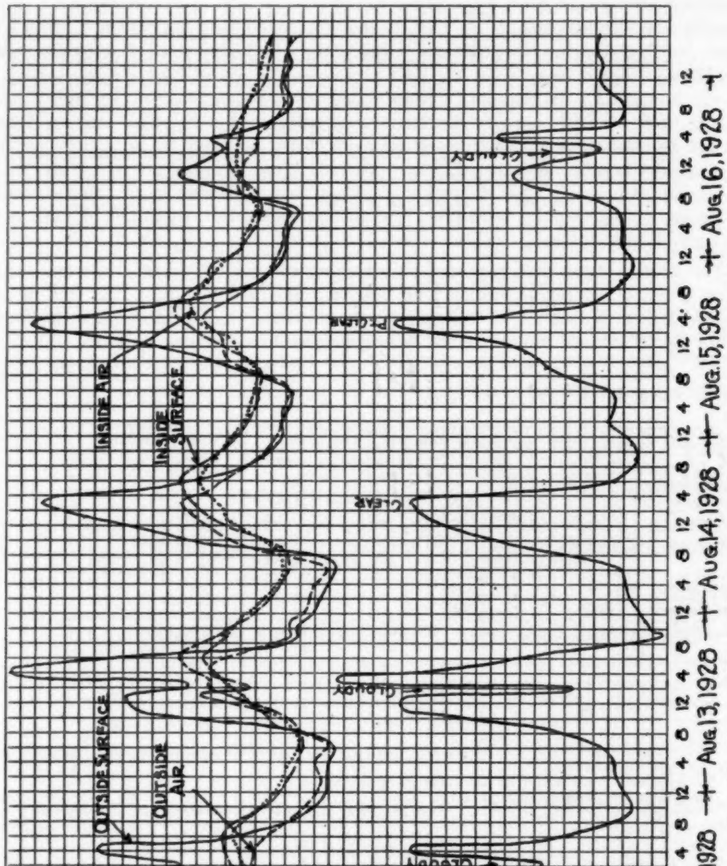


Fig. 12. Log of Heat Flow through Aluminum Shingle Roof

ing transmittance coefficients from the component film conductance and conductivity coefficients.

The location of the section of basement floor, through which the heat flow was measured, in relation to the outside wall and fill is shown in Fig. 6. Temperatures were observed 6 in. above the floor, at the surface of the floor, (under

the 1.5 in. thickness of concrete) at a point in the earth 11.5 in. below the surface, and also 19.5 in. below the surface of the floor.

Fig. 11 is a log of heat flow through the floor and the accompanying temperatures. It will be observed that the earth temperature either 11.5 in. or 19.5 in. below the surface of the floor varied little, and also that the variations in the temperature of the basement air were not great. The average air temperature throughout the period of the test in the basement 6 in. from the floor was 65.7 degrees with a maximum variation of $\pm$ 1 deg. Perhaps the best single value expressing heat loss through the floor is the average rate of heat flow of 4.33 Btu per hour per square foot for an average temperature; difference of 10.5 degrees between basement air and the earth 19.5 in. below the surface of the floor. This gives a heat flow of 0.412 Btu per hour per square foot per degree Fahrenheit; difference between air and earth 19.5 in. below floor.

The thickness of the concrete floor, about 1.5 in., and the cinder fill, about 2 in., were not uniform enough to allow accurate determinations of conductivity of these materials. This may account for the high conductivity coefficient, $k = 2.32$ Btu per hour per square foot per degree Fahrenheit; difference per inch thickness, found for the cinder fill compared with a conductivity, $k = 0.70$ for a cinder fill in a roof previously reported by the Laboratory. The difference in dryness of the cinders in the roof and those under the concrete floor may, however, account for the discrepancy. The earth under the floor was a reddish clay damp enough to be slightly plastic. It had a conductivity of $k = 8.46$ Btu per hour per square foot per degree Fahrenheit; difference per inch thickness.

The aluminum shingle roof was tested under summer conditions when the heat flow was inward during the day and outward at night. The coefficient given in Table 1 are averages for day and night. As previously found in tests made in hot weather, the outside film conductance f_o is negative. Fig. 12 is a log of heat flow through this roof.

Conclusions

The coefficients as given in Table 1 verify the computed values contained in THE GUIDE in most instances. Where the coefficients found are not in agreement with accepted values, the discrepancies usually can be accounted for by variation in construction. The conductivity of concrete is the only case showing marked discrepancy between measured and computed values, indicating that the coefficients used in the past may be too low and that heat flow through concrete walls of different mix and aggregate should be the subject of further study.

DISCUSSION

H. P. REID (WRITTEN): In commenting on this paper, I wish to limit my remarks to that part of the paper which refers to the heat flow through concrete structures. There are two tests reported on such strucures, one giving the heat flow and corresponding coefficients through a 12-in. concrete block wall and the other the transmittance through and heat flow curves for a 1½-in. concrete basement floor with a cinder fill under this concrete.

Nothing is given as to the age of the concrete in either of these tests, whether it was comparatively new or whether it had been built several years. From such information as we have to date apparently the heat loss through concrete is much

greater the first few months after the concrete has been poured than it is several years later.

The values for k as determined by these tests and the various values reported in THE GUIDE vary so widely as to indicate the desirability of much further research into the heat flow through concretes. As an illustration, in THE GUIDE, 1929, the values of k for a stone concrete with a 1-2-4 mix is given by one authority as 8.30 while another authority gives a value of 6.27 for a similar concrete except that it is made of a 1-2-5 mix. It is hardly conceivable that this change in mix could alone account for this great difference in thermal conductivity. The value for k as given for the concrete block wall in this paper is 7.08, another value between the two above.

In looking up such tests as I have been able to find in the short time allotted, I find values for k for various concretes that vary over the range from 0.483 for a cellular concrete having a specific gravity of 0.25 to 16.6 for a 1-5.6-5.1 dolomite pebble aggregate concrete with a specific gravity of 2.16. Again Dr. Henky of Munich reports the value of k for a gravel structural concrete as 5.62.

It appears that the heat of conductivity of concrete is primarily a function of:

(*a*) Ratio of solid material to total void space in the concrete.
(*b*) Specific gravity of the concrete.
(*c*) Free moisture of the hardened concrete.
(*d*) Temperature at which tests are made.

Therefore engineers should not use a single value of k for all concretes for their transmittance computations, but this value must vary with the nature of the specific concrete of the structure in question. Furthermore, wherever possible the heating engineer should specify the nature of the aggregate to be used and the mix desired where such specifications will give the desired structural strength.

It is being found practical to *insulating* concretes. This insulated structure is being particularly applied to floors and to roofing tile. I recently had occasion to observe the replacement of a rotted wood floor which had been laid on concrete, the latter being laid on sand. The wooden floor had been cold and damp even though there was about 1 in. of air space between it and the base concrete. In the reconstruction an asphalt tar was painted over the concrete base, and approximately 1 in. of mortar insulation was laid on this. After drying the insulation, tar paper was placed on it and about 1½ in. of concrete surface floor poured. The same office force was put back to work on this flooring with no covering over the concrete. They report a very comfortable floor all this winter while they had continually complained of discomfort on the wood over concrete.

For roofing tile recently laid over a new steam turbine in a power house a tile made of Portland cement concrete with 1 in. of insulation as core was used. This turbine bleeds a small amount of steam from the bearing seal to the atmosphere of the power house whenever operated. While this turbine has not been placed into regular operation, yet it has been turning over on its own steam for the past two weeks or so and no noticeable ceiling sweating has resulted.

Let me again urge the need of further research into the heat losses through concretes.

PROF. J. D. HOFFMAN: I am very glad to hear the explanation given by Mr. Houghten, regarding the heat transmission through a brick veneer wall.

This happens to be one of the most common types found in ordinary residence construction and I have been concerned to know how the difference between the value in THE GUIDE and the theoretical value as ordinarily calculated came about.

H. M. HART: I may not be up-to-date, but what I wanted to ask is, When is an air space dead?

We speak of these air spaces, but I am somewhat reluctant sometimes to consider an air space a dead air space, because I am quite sure it is going to be a leaky air space, and I wonder if we can call it a dead air space in those cases where we are sure it can still breathe. It may be said that blocking off may increase the value of resistance to an air space but what distance?

PROF. F. B. ROWLEY: The limiting ratio between area and thickness of air space is a subject of further investigation. All of the air spaces which were tested by the hot plate method were 9 sq in. and those by the hot box method were 3 sq ft. The thickness ranged from 0.1 up to 0.75 in. for the hot plate tests and from 0.50 in. up to 3.5 in. for the hot box tests. These ratios are so large that there seems to be no question of limiting the circulation.

The question of leakage of air into or out of the air space is certainly an important one. If the material inclosing the space is porous, some of the value of the air space is lost as the apparent conductivity is increased. Therefore, in considering the air spaces, they are considered as impervious to the passage of air.

In speaking of Mr. Houghten's paper, and also Mr. Reid's discussion, I am glad to hear a man, who is interested in cement and concrete, say that there ought to be more research along this line. There is no question but what there should be more experimental data to determine the thermal resistance of the various aggregrates and mixtures, time of setting, etc.

In regard to Mr. Houghten's paper it seems to me he inferred that he was closing up the work for the heat meter; one of the valuable pieces of work with the heat meter was shown in the last curve of his paper. The heat meter is an ideal method of finding out what is going on in a wall, due to the heat from the sun, the wind, etc., on the outside. These are conditions which cannot be gotten in the laboratory, and whether or not the heat meter checks exactly with calculated constant, it does give us the relative conditions and shows what is going on from day to day with different outside weather conditions. It provides a practical solution that cannot be duplicated in the laboratory.

L. A. HARDING: May I say a word on the Laboratory paper? There is one thing in regard to Director Houghten's paper that may have escaped the average reader, and I think it is well to point it out, inasmuch as we contemplate the discontinuation of these tests at the Pittsburgh Laboratory.

The heat flow curves bring out one feature and that is up to the present time we have no practical method or scheme devised whereby we can correct the estimated heat transmission of a wall or roof, due to its heat capacity. The difficulty is that we have continually changing outside temperature conditions.

It is a well known fact, if you have a thick wall that has a high heat capacity, it will tend to iron out the heat flow, and you can probably get by with a less amount of radiation than you could with a thin wall that has the same conductivity but with a smaller heat capacity.

There was one statement made by Mr. Reid that I do not want to let go unchallenged. I do not believe the heating engineer will ever be able to specify the aggregate for concrete. The specifying of the concrete aggregate is not within the province of the heating engineer; but rather the province of the structural engineer.

F. C. HOUGHTEN: As pointed out by Mr. Hart and Professor Rowley, the dimensions and tightness of an air space in a compound wall must be taken into consideration in determining its resistance to flow of heat. It is certainly not fair to determine the insulating value of a perfectly tight air space in a laboratory and apply this value to practical walls. The air space between the sheathing and the brick of a brick veneer wall has many openings to the outside through which air can leak, as demonstrated by the Laboratory study of air leakage through brick walls. It would seem that a factor for such an air space must be determined under natural conditions of leakage in order to apply it to such walls.

Answering Mr. Reid, the concrete floor and the concrete block basement wall reported in the paper were in a house which had been built about one year.

In an earlier Laboratory report on this same subject there were given data on change in conductivity of concrete with aging. These data showed that the conductivity of concrete decreased considerably during the first several months after pouring. Additional data have since been obtained on this same block showing that its conductivity has continued to decrease; about a year after pouring, the conductivity of the concrete was still considerably higher than that used in THE GUIDE showing a value of approximately 14, whereas THE GUIDE coefficients are based on a conductivity of 8.30. While a change in conductivity a year after pouring was apparently still taking place the rate of change had greatly decreased and it is doubtful if continued aging will reduce the conductivity of this slab much below the present value.

As pointed out by Mr. Reid the conductivity of concrete apparently varies greatly with the mix and nature of the aggregate. It is quite probable that with the same mix, the conductivity of concrete would differ greatly depending upon the nature of the aggregate—that is, if the pebbles or crushed stone were largely granite, limestone, or some other form of rock.

It is apparent that such variables may change the results obtained, from the value of 8.30 appearing in THE GUIDE or even lower as obtained by some investigators, to 10 or 14 as found in some of the Laboratory determinations, indicating that there is still need for considerable investigation of this subject.

Mr. Reid apparently misinterpreted the conductivity value of the concrete block reported in the present paper. The value, 7.08 reported, was the apparent conductivity of the block including the air spaces and not the conductivity of the concrete itself which would of course be considerably higher.

No. 825

THERMAL RESISTANCE OF AIR SPACES

The results of cooperative research between the University of Minnesota and the American Society of Heating and Ventilating Engineers

By F. B. Rowley[1] (*Member*), and A. B. Algren[2] (*Non-Member*)
Minneapolis, Minn.

THERE are two generally accepted methods of measuring the flow of heat through building materials, the hot-plate and the hot-box methods. No detailed discussion of either method is necessary at this time as both have been thoroughly outlined in previous articles published in the Journal of the American Society of Heating and Ventilating Engineers. While there are advantages and disadvantages for each method, it may, in general, be said that the hot plate is best adapted for determining the thermal conductivity of homogeneous materials and the hot box for determining the coefficient of conductivity for built-up wall sections, that is, the wall sections which are composed of various materials combined with air spaces. The hot plate gives the conductivity from outside surface to outside surface of the material, while the hot box gives the conductivity from air on one side to the air on the other side of the wall. The latter is the constant usually required for practical purposes, but the former is the most easily obtained and the one which is generally used to give the thermal value of materials.

Theoretically it is possible to convert the values obtained by one method to those obtained by the other by the following formulae:

$$U = \frac{1}{1/K_1 + X_1/C_1 + 1/K_2 + X_2/C_2 + \text{etc.}}$$

in which

U = the coefficient of conductivity from air to air

K_1, K_2, etc. = the surface coefficients

X_1, X_2, etc. = the thickness of the various materials used

C_1, C_2 etc. = the conductivity of the respective materials.

In practice the results obtained by these conversions have been questionable, due to the lack of knowledge concerning the value of the surface coefficients.

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Surface Factors

Surface factors may be divided into two groups; those on the outside surfaces of the built up walls and those on the interior surfaces. The purpose of the present research has been to determine the values of the interior coefficients under different conditions. In order to eliminate the question of proper air temperatures, the surface coefficients on each side of the air space together with

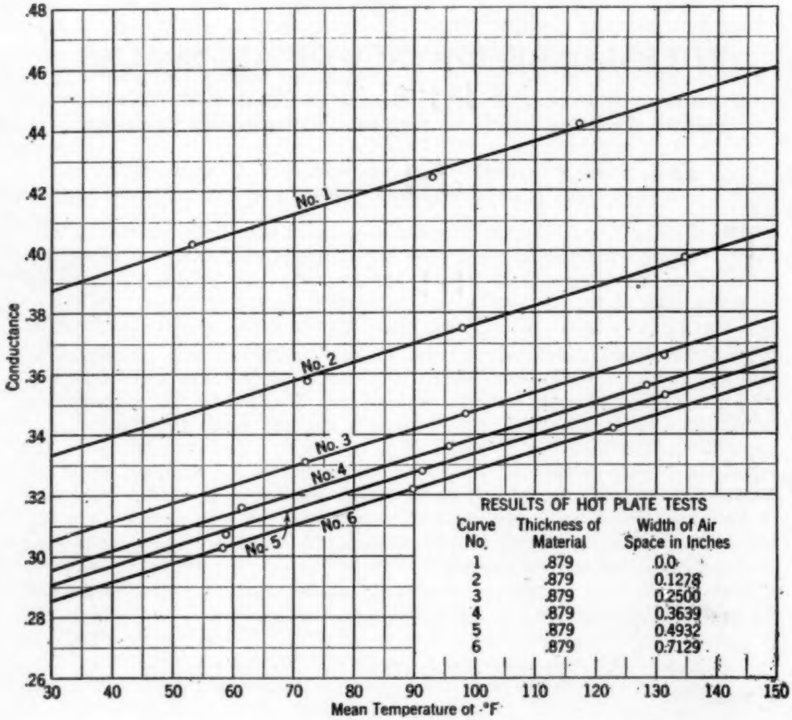


FIG. 1. CONDUCTANCE OF MATERIAL *A* WITH AND WITHOUT AIR SPACES

the air resistance has been treated as one factor, and is described here as the resistance or the conductance, as the case may be, from surface to surface of the air space. The tests were made on both the hot-box and hot-plate apparatus as described in the paper entitled, "Heat Transmission Research" given at the Semi-Annual Meeting of the Society in the spring of 1928 and published in the July, 1928 issue of the JOURNAL. No further description of this apparatus should be necessary.

Factors Affecting Air Space Coefficients

There are several factors which affect air space coefficients. These are width of air space, mean temperature between surfaces, condition of surfaces and the ratio of the area to thickness of air space. The effects, due to width of space and mean temperature, should be the same for all surfaces. The surface effects will be different for different classes of materials, although in many cases it

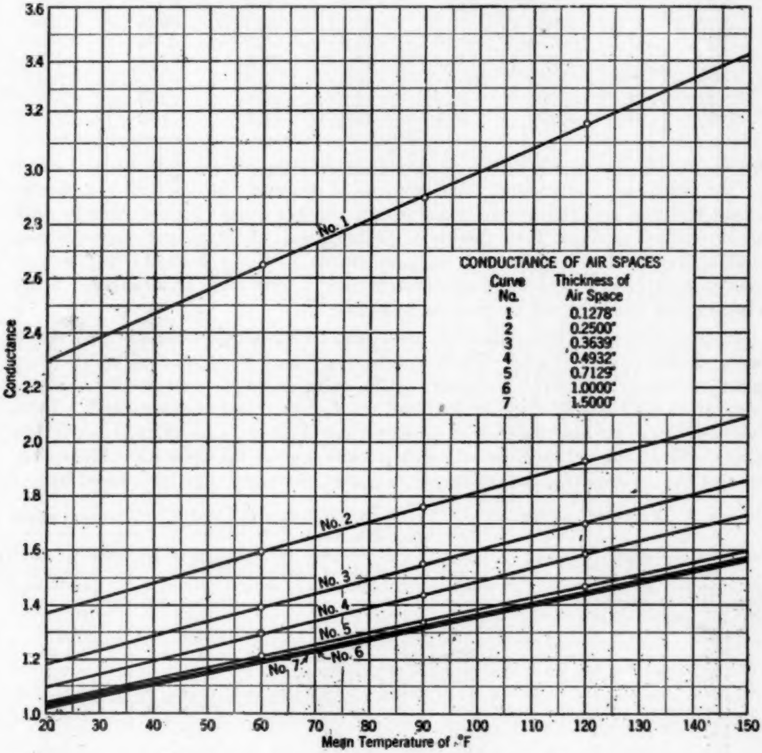


FIG. 2. CONDUCTANCE OF AIR SPACES FROM FIG. 1

should be possible to group materials having similar characteristics. The effect of ratio of surface area to thickness will be more pronounced for small ratios, and it is probable that for average building constructions it may be neglected.

In these experiments the air space coefficients were determined for different width of air space and different mean temperatures. The surfaces used were Insulite, Masonite, Flaxlinum, Celotex Compo board and Gypsum board, the last two being paper covered. The limiting areas of the air spaces were 9 in. square for the hot plate and 36 in. square for the hot box. As will be shown

TABLE 1. CONDUCTIVITY OF MATERIALS *A, B, C, D* AND *E*—HOT-PLATE METHOD
Used for determining air-space coefficients

Test No.	Material	Thickness	Temp F High Side	Temp F Low Side	Mean Temp.	Conductance	Conductivity per Inch
144B	*A*	0.8791	60.6	45.6	53.1	0.403	0.353
146A	*A*	0.8791	138.8	47.2	93.0	0.424	0.373
144A	*A*	0.8791	187.8	46.9	117.3	0.440	0.387
152A	*B*	1.0520	116.7	40.8	78.8	0.345	0.363
62A	*C*	0.4820	78.25	42.4	60.3	0.746	0.359
72	*C*	0.4754	102.0	43.8	72.9	0.782	0.372
72A	*C*	0.4782	146.7	45.2	95.95	0.794	0.379
72B	*C*	0.4774	201.1	46.6	123.8	0.823	0.393
14	*D*	0.5201	78.38	42.9	60.6	1.220	0.634
149	*E*	0.7280	81.9	42.9	62.4	1.650	1.20

by the results, the thickness of space and mean temperature both had an effect on the air-space conductance. The surfaces of the materials used all had similar characteristics and the air-space values on the hot plate checked exactly with those on the hot box. It is, therefore, probable that the data and curves shown will apply to all materials with surface similar to those specified and for air spaces used in practical building construction. Additional materials are being checked but data are not, at present, available.

In performing the tests, sheet of homogeneous materials were selected which were uniform in thickness. Their conductivity was first obtained by the hot-plate method after which two sheets of the material were separated by skeleton separators to a given distance and the conductivity again determined. Tests were made at various mean temperatures and with various thicknesses of air space. Since the only difference in the conditions for the tests was the separation of the material for the different air spaces, the resistance introduced by the

TABLE 2. RESULTS OF HOT-PLATE TESTS—TWO THICKNESSES OF MATERIAL WITH AIR SPACE BETWEEN

Test No.	Material	Total Thickness of Material	Width of Air Space, In.	Mean Temp F	Conductance
121	*A*	0.8791	0.128	72.2	0.358
120	*A*	0.8791	0.128	98.0	0.375
119	*A*	0.8791	0.128	134.9	0.398
122	*A*	0.8791	0.250	72.0	0.331
124	*A*	0.8791	0.250	98.6	0.347
125	*A*	0.8791	0.250	131.4	0.366
132	*A*	0.8791	0.364	61.4	0.316
131	*A*	0.8791	0.364	95.9	0.336
130	*A*	0.8791	0.364	128.4	0.356
137	*A*	0.8791	0.493	58.9	0.307
138	*A*	0.8791	0.493	91.3	0.328
139	*A*	0.8791	0.493	131.6	0.353
140	*A*	0.8791	0.713	58.3	0.303
141	*A*	0.8791	0.713	89.9	0.322
142	*A*	0.8791	0.713	122.9	0.342
152	*B*	1.0520	0.252	79.3	0.286
144	*D*	0.5201	0.493	69.8	0.630
153	*C*	0.9179	0.250	74.7	0.320
154	*D*	0.5201	0.051	60.2	1.045
155A	*E*	0.7280	0.250	74.45	0.825

air space was readily calculated. By this method air space coefficients were obtained for spaces varying from 0.051 in. up to 0.713 in. by the hot plate and from ½ in. to 1½ in. by the hot box. The separators were built of pine strips ⅛ in. wide for the hot plate and about ¾ in. for the hot box; thus the area occupied by the strips was very small as compared to the air-space area. Thus

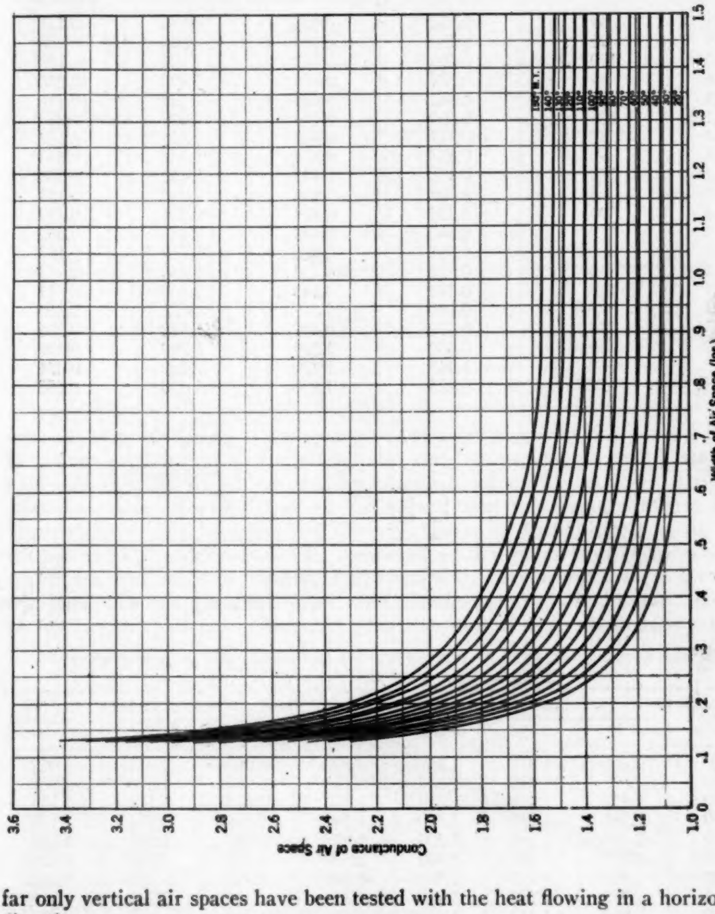


Fig. 3. Conductance of Air Spaces for Different Mean Temperatures Fahrenheit

far only vertical air spaces have been tested with the heat flowing in a horizontal direction.

Heat is transmitted through an air space by radiation, conduction and convection. Radiation depends upon the nature of the surfaces and their temperatures. It is independent of the width of the air gap. For air without motion the heat transmitted by conduction is inversely proportional to the thickness

TABLE 3. CONDUCTANCE OF AIR SPACES BETWEEN SURFACES OF MATERIALS, *A, B, C, D* AND *E*—BY HOT-PLATE METHOD

Values for material derived from the curves of Fig. 1. All other values derived from Tables 1 and 2

Material	Mean Temp F	Conductance Material and Air Space	Conductance of Material	Conductance of Air Space	Width of Air Space, In.
A	60	0.352	0.406	2.646	0.128
A	90	0.370	0.424	2.896	0.128
A	120	0.389	0.443	3.168	0.128
A	60	0.324	0.406	1.592	0.250
A	90	0.342	0.424	1.759	0.250
A	120	0.360	0.443	1.925	0.250
A	60	0.314	0.406	1.386	0.364
A	90	0.333	0.424	1.545	0.364
A	120	0.351	0.443	1.693	0.364
A	60	0.309	0.406	1.297	0.493
A	90	0.328	0.424	1.437	0.493
A	120	0.346	0.443	1.583	0.493
A	60	0.304	0.406	1.210	0.713
A	90	0.322	0.424	1.336	0.713
A	120	0.340	0.443	1.468	0.713
B	78.8	0.286	0.345	1.672	0.250
C	74.7	0.320	0.402	1.570	0.250
D	69.8	0.630	1.229	1.292	0.493
D	60.2	1.045	1.220	7,280	0.051
E	74.45	0.825	1.650	1.650	0.250

of air space. The amount transmitted by convection is dependent upon the temperature difference between the two sides of the air space and the freedom of the air to circulate. Thus for the same surfaces, as the air space is increased from zero, the amount of heat transmitted by radiation will remain constant. The amount transmitted by conduction will be decreased and the percentage transmitted by convection will be increased. As an air space is increased the decrease in heat transmitted by conduction is greater than the increase in the amount transmitted by convection. This ratio gradually changes until the greater part of the heat is transmitted by convection and that transmitted by

TABLE 4. CONDUCTANCE OF AIR SPACES AT DIFFERENT MEAN TEMPERATURES

Values taken from the curves of Fig. 2

Mean Temp. F	Conductance of Air Spaces for Various Widths in Inches						
	0.128	0.250	0.364	0.493	0.713	1.00	1.500
20	2.300	1.370	1.180	1.100	1.040	1.030	1.022
30	2.385	1.425	1.234	1.148	1.080	1.070	1.065
40	2.470	1.480	1.288	1.193	1.125	1.112	1.105
50	2.560	1.535	1.340	1.242	1.168	1.152	1.149
60	2.650	1.590	1.390	1.295	1.210	1.195	1.188
70	2.730	1.648	1.440	1.340	1.250	1.240	1.228
80	2.819	1.702	1.492	1.390	1.295	1.280	1.270
90	2.908	1.757	1.547	1.433	1.340	1.320	1.310
100	2.990	1.813	1.600	1.486	1.380	1.362	1.350
110	3.078	1.870	1.650	1.534	1.425	1.402	1.392
120	3.167	1.928	1.700	1.580	1.467	1.445	1.435
130	3.250	1.980	1.750	1.630	1.510	1.485	1.475
140	3.340	2.035	1.800	1.680	1.550	1.530	1.519
150	3.425	2.090	1.852	1.728	1.592	1.569	1.559

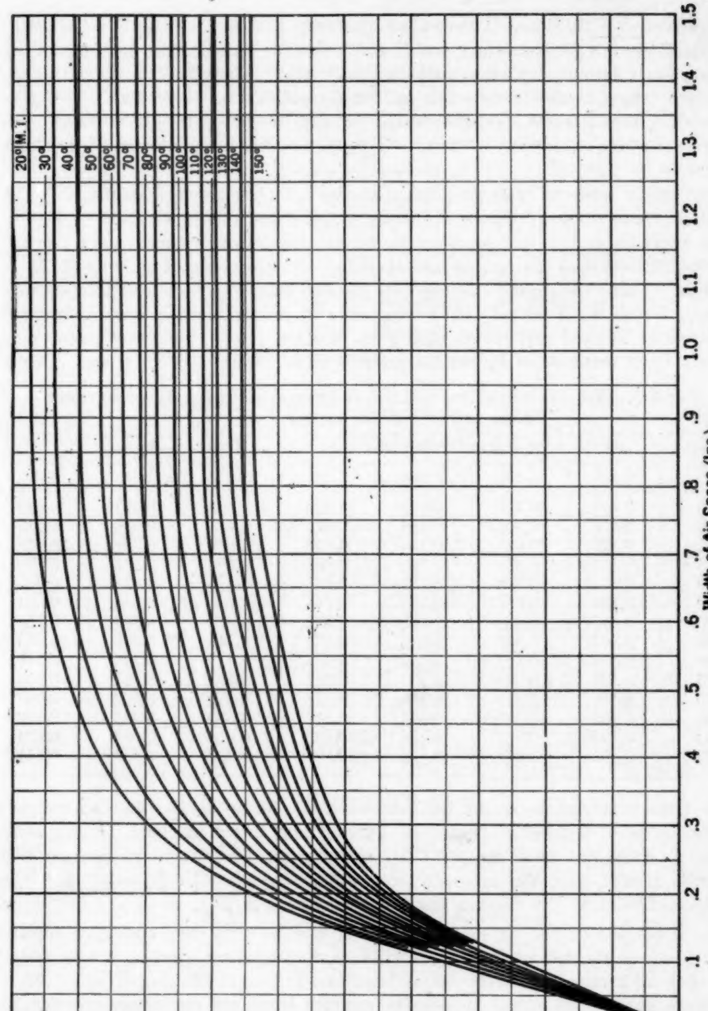


FIG. 4. RESISTANCE OF AIR SPACES FOR DIFFERENT MEAN TEMPERATURES FAHRENHEIT

conduction becomes a negligible factor. This applies only to vertical air spaces and the combined effect is very evident when referring to the curves of Figs. 3 and 4.

In these tests six different materials were used and were designated by *A*, *B*,

C, D, E and *F*. *A, B* and *C* were fibre insulating boards. *D* and *E* were building boards with a paper surface and *E* was a flexible flax board. In some of the hot-box tests lath and plaster surfaces were also included. The results will, therefore, apply to such materials as Insulite, Celotex, Masonite, Flaxlinum, Fibrofelt, Compo board, Gypsum board or any of the materials covered with paper or of similar fibrous surfaces. The results for other classes of surfaces remain to be checked.

A complete series of tests was run with the hot plate using material *A*. The results of these tests are shown in Tables 1 and 2 and the curve sheet of Fig. 1. From curve sheet 1, it will be noted that mean temperature curves were run for the material without air spaces and then for air spaces ranging from 0.128 to 0.713 in. These tests gave substantially parallel lines for the mean temperature curves. From these curves the conductance of the air spaces were calculated as shown in Table 3 and curves of Fig. 2. Curves 1 to 5, inclusive, were taken directly from material *A* by the hot-plate method, while Curves 6 and 7 were

TABLE 5. RESISTANCE OF AIR SPACES AT DIFFERENT MEAN TEMPERATURES
Values taken from the curves of Fig. 2

Mean Temp F	Resistance of Air Spaces for Various Widths in Inches						
	0.128	0.250	0.364	0.493	0.713	1.00	1.50
20	0.435	0.730	0.847	0.909	0.962	0.971	0.978
30	0.419	0.702	0.810	0.871	0.926	0.935	0.939
40	0.405	0.676	0.776	0.838	0.889	0.899	0.905
50	0.391	0.651	0.746	0.805	0.856	0.868	0.870
60	0.377	0.629	0.719	0.772	0.826	0.837	0.842
70	0.366	0.607	0.694	0.746	0.800	0.806	0.814
80	0.355	0.588	0.670	0.719	0.772	0.781	0.787
90	0.344	0.569	0.646	0.698	0.746	0.758	0.763
100	0.334	0.552	0.625	0.673	0.725	0.734	0.741
110	0.325	0.535	0.606	0.652	0.702	0.713	0.718
120	0.316	0.519	0.588	0.633	0.682	0.692	0.697
130	0.308	0.505	0.571	0.613	0.662	0.673	0.678
140	0.299	0.491	0.555	0.595	0.645	0.654	0.658
150	0.292	0.478	0.540	0.579	0.628	0.637	0.641

taken from material *E* by the hot-box method. Material *E* was also checked by the hot-plate method as shown in Table 3 and found to check exactly with material *A* by the same method. It is therefore, reasonable to assume that material *E* will give the same air space characteristics as material *A*. The results obtained by the hot-box method are shown in Fig. 5.

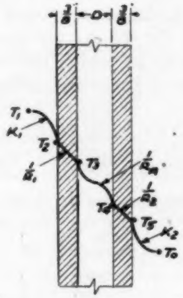
An inspection of the curves of Fig. 2 will show that they converge toward some point at the left of the origin. By extending these curves, it was found that they all crossed the zero conductance line at points ranging from —207 to —247 F. While these are apparently straight lines for the range covered by the tests, it is probable that they would not continue so when indefinitely extended, and it cannot be assumed that the conductance would be zero at these temperatures. While the complete set of curves were derived from material *A*, checks of several points were made by the other materials listed and in several cases check tests were run by both the hot-plate and the hot-box methods.

In order to get the conductance and resistance of air spaces for other thicknesses than those tested and for various mean temperatures, the curves of Figs.

4 and 5 were plotted directly from the curves of Fig. 2. The data from which these curves were plotted are given in Tables 4 and 5. The points of Table 4 were taken directly from the curve for the mean temperatures indicated and those for Table 5 are the reciprocal values. Since the mean temperature curves of Fig. 2 are straight lines, curves might have been drawn on Figs. 3 and 4 for a greater range of mean temperatures. These conditions are, however, being

TEST SECTION

Material E



U = Coefficient of Transmission Air to Air.

$\frac{1}{R}$ = Conductance of Material E

K = Surface Constant

T = Temperature — Degrees F.

R_A = Thermal Resistance of Air Space

$\frac{1}{R_A}$ = Conductance of Air Space

RESULTS OF TESTS										
D	½ Inch				1 inch				1½ inches	
	Run 1		Run 2		Run 1		Run 2		Run 1	
	Const	M.T.	Cons't	M.T.	Cons't.	M.T.	Const	M.T.	Cons't	M.T.
U	.370	39.68	.404	59.91	.365	40.11	383	60.12	.361	40.07
$\frac{1}{R_1}$	3.428	58.65	3.556	67.77	3:614	58.65	3.684	69.10	3.388	58.47
$\frac{1}{R_2}$	3 508	24.05	3.173	60.35	3.290	24.15	3.516	51.60	3.310	23.82
K_1	1.658	70.99	1.639	74.98	1.667	71.48	1.690	75.77	1.664	71.45
K_2	1.460	9.58			1 484	9.86	1.628	44.65	1.469	9.72
$\frac{1}{R_A}$	1 193	40.80	1.284	59.20	1.126	41.60	1 172	60.40	1.114	41 20
R_A	838	40.80	.779	59.20	.888	41.60	.857	60.40	898	41.20
TEMPERATURES										
T_o	-.63		39.9		-.03		39.9		0.0	
T_1	79 98		79 92		80.26		80.35		80.14	
T_2	62.0		70.05		62.7		71.2		62.75	
T_3	53.3		65 5		54.6		67.0		54.20	
T_4	28.3		52.9		28 6		53.8		28.20	
T_5	19.8		47.8		19 7		49.4		19.45	

FIG. 5. RESULTS OF HOT-BOX TESTS FOR DIFFERENT AIR SPACES

checked with further tests and will be included in a later report. Referring to Fig. 3, it will be noted that the conductances are increasing very rapidly at the last point taken which corresponds to Curve 1 of Fig. 2 or an air space of 0.128 in. Theoretically all of these curves extended should become tangent to the zero air space line at infinity. In order to get the points in this region of small air-spaces, curves of Fig. 4 were plotted. Theoretically all curves for air space resistance should pass through the zero point as the resistance of a zero air space will be zero. Some question might be raised as to the accuracy

of this assumption, due to the fact that the air space as taken is a combination of surfaces and air space and that with zero air space there would still be the surface resistances. However, all of the curves drawn through the tested points were directed toward the zero point. In all cases the curves include points of air spaces for widths as low as 0.128, and in the case of the 60 F mean temperature, a test was made with an air space of 0.051 in. This point checked exactly with the curve as drawn. While some question may be raised as to the point passing exactly through zero, the curves would indicate that this is the case and that that the real surface resistance is a very small factor as compared with the air space resistance. As the curves are extended to the thicker air spaces, they approach horizontal lines. There is no definite point at which the maximum resistance of the air space is reached, but apparently the real effectiveness ends at from 0.7 to 0.8 in.

In previous work several walls tested by the hot-box method have been provided with thermocouples placed on the inside surfaces and data have been taken

TABLE 6. CONDUCTANCE OF AIR SPACES IN WALLS TESTED BY THE HOT-BOX METHOD

Wall No.	Surfaces	Width of Air Space, In.	Mean Temperature	Conductance
23	*C* to *C*	3.625	41.2	1.011
23	*C* to *C*	3.625	37.0	1.009
25	*B* to *B*	3.625	42.2	1.010
27	*B* to *B*	3.625	47.6	1.075
30	*B* to *B*	3.625	47.3	1.087
30	*B* to *B*	3.625	47.4	1.098
30	*B* to *B*	3.625	101.1	1.362
30	*B* to *B*	3.625	102.2	1.325
34	Paper to wood	1.440	37.25	0.922
34	Paper to wood	1.440	36.8	0.910
36	Paper to *E*	1.440	67.9	1.157
36	Paper to wood	1.440	39.3	1.028
36	Paper to *E*	1.440	67.1	1.144
36	Paper to wood	1.440	36.9	1.002
32	Lath and plaster to *F*	1.440	63.3	1.103

to determine the conductance of the air spaces. These spaces have varied in width from 1.4 in. to 3.62 in. and the mean temperatures have varied from 36.9 F to 102.2 F. The surfaces have been composed of such materials as Insulite, Flax-li-num, Celotex, Gypsum board, lath and plaster, wood and paper. The air space coefficients obtained from these various tests are shown in Table 6. When these are plotted on Fig. 2, it is found that they range slightly below Curve 7. This does not necessarily mean that the air spaces show a larger resistance. It may be explained by the fact that the walls were built up of 2 x 4 studdings placed 16 in. on centers. Two of these studs passed down over the three-foot test area and formed a resistance over a part of this area which is credited to the air space. The significance of this is apparent when it is noted that the area covered by the studs is approximately 0.75 of a square foot or 8.3 per cent of the test area, and the resistance of each stud per unit area is about three times that of air space per unit area. The studding decreases the heat transmission through the test wall, and since the conductance of the air spaces was calculated from this conductance, they will be somewhat lower than those obtained with walls having no obstructions in the air space.

The results of this investigation show that the air space coefficients are the same for many materials, and undoubtedly for practical purposes, all materials used in building construction may be classified into a small number of groups, with characteristic constants for each group. With these constants and the proper external surface constants, it will be possible to calculate with accuracy the overall coefficients of heat transmission from the hot-plate conductivities.

DISCUSSION

P. D. CLOSE (WRITTEN): The hot plate tests reported in this paper indicate conductances of air spaces approaching unity, whereas the results of the hot box tests indicate values which approximate 1.0 for spaces ranging from 1.4 in. to 3.62 in. The air space curves presented in this paper are asmytotic as the width of the space increases so that beyond a certain thickness no additional increase in insulating value is obtained with an increase in the width of the air space. For practical purposes it can probably be assumed that an average vertical air space 1 in. or more in width has a conductance of 1.0 or 1.10.

A 1-in. thickness of an insulation having a conductivity of 0.25 installed in such a way as to create an additional air space would therefore have an effective thickness of 1¼ in. as installed. On the other hand, a 3⅝ in. thickness of a fill having a conductivity of 0.50 would have an effective thickness as installed of 3⅛ in. since it would require a ½ in. thickness of a material of this conductivity to equal an average air space of an inch or more in width, based on the assumption that the conductance of an air space is 1.0.

The opinion has prevailed for some time that an air space possesses a high degree of heat resistance. Operative builders will frequently tell you that the buildings they are offering for sale are well insulated because the walls contain *dead* air spaces. The public in general has little knowledge of the relative insulating values of air spaces, commercial insulations and building materials.

Some concerns have been inclined to exploit the added heat resistance obtained in walls by means of air space construction, whereas others have shown a tendency to discount any insulating value thereby obtained.

For example, some organizations identified with the marketing of felted insulations are commercializing on the idea that by the use of their materials installed in the manner recommended by them, the insulating value of an extra air space is obtained in addition to that of the material itself.

Another class of materials—*fills*—cancel an air space by the manner in which they are usually installed, the exception being where only a part of the air space is filled.

The third type of insulation—*the board form*—is installed in such a manner in most cases that the number of air spaces in the construction is not changed, although it is also possible to apply this type of insulation so that additional air spaces are obtained.

Costs of materials and their application vary. Moreover, the proper economic thickness of insulation will vary with the type of heating system installed and the kind of fuel burned, so that it is difficult to obtain an intelligent comparison between materials of different types on the basis of their thicknesses and con-

ductivities alone, even giving consideration to whether the number of effective air spaces is changed by the use of the materials under consideration. A true comparison can only be obtained on the basis of the over-all transmissions, and fuel, radiation and construction costs, giving due consideration to quality of materials.

The surface factors used in THE GUIDE 1929 for computing heat transmission coefficients of walls and roofs are based on tests conducted at the *University of Illinois* and *Penn State College*. These values have for some time been regarded as the most reliable and authentic available and give an average air space conductance of 0.67 for so-called *still* air, or an average surface conductance of 1.34.[8]

The air space conductances presented in the paper under discussion are somewhat higher than are obtained by the use of the surface coefficients used in THE GUIDE. Other recent tests would also indicate that surface and air space coefficients in most cases are higher than those now in use. The Bureau of Standards letter circular 227 states that an average air space more than 1 in. wide is equivalent to about ¼ in. of insulating material, although no reference is made to the conductivity of the air space upon which this comparison is based.

More recent tests on air spaces conducted at the Bureau of Standards apparently substantiate an air space value of approximately this magnitude. Surface coefficients based on tests conducted at Armour Institute a few years ago gave values for several materials averaging 2.18, which is equivalent to an air space conductance of 1.09, and which compares favorably with the data submitted in the paper under discussion. Other tests conducted at the *Laboratory for Technical Physics* and the *Research Institute for Heat Conservation* in Germany, as well as by Professor Bugge, a Norwegian experimenter, show air space conductances ranging between those reported in the paper under discussion and those obtained by using the surface coefficients in THE GUIDE 1929. The *University of Illinois* Engineering Experiment Station Bulletin No. 102 reports air space coefficients ranging from 1.0 to 1.7, compared with the assumed average value of 1.0 based on the University of Minnesota tests reported in this paper and 0.67, the value used in THE GUIDE 1929. The *A.S.R.E.* have arbitrarily recommended a surface resistance of 0.5 for some time, but have made no allowance for air movement over outside exposed surfaces. This surface resistance value of 0.5 is equal to an air space conductance or resistance of 1.0, the value derived from the results reported in the paper under discussion.

There seems to be sufficient reliable test data available to indicate that the correct value for an average vertical air space of 1 in. or more in width is about 1.0 for ordinary temperatures. It would be possible from the data published in this paper to compute the heat transmission through any type of wall construction and assign the proper air space coefficient for almost any width air space used in ordinary building construction. The accuracy involved, however, does not warrant this degree of refinement and it is probably sufficient to assign one air space or surface factor to all air spaces beyond a certain width and to neglect all air spaces less than this width. If it is assumed that the resistance of a surface exposed to a wind velocity of 15 mph is 1/3 that of a surface in still air (as has been the practice in computing the over-all coefficients in THE

[8] An air space conductance of 1.10 was used in THE GUIDE, 1930, based on the results of this paper.

Guide) the resistance of such outside surfaces is so small compared with the total resistance of the wall that it may be safely neglected in the majority of cases. The average resistance of an outside surface based on a still air surface resistance of 0.5 is 0.167, and in the average wall construction the difference between a still air resistance of 0.5, a 15-mile wind velocity resistance of 0.167 and no resistance (that is, neglecting the outside resistance entirely) is so small that it undoubtedly is sufficiently accurate to base transmission coefficients on one wind velocity, rather than to attempt to allow for any other prevailing wind velocity than the one on which the overall coefficient was based.

In the preparation of future editions of The Guide it is proposed that consideration be given to the data published in the paper under discussion as to the advisability of changing the surface factors used.

Armin Elmendorf (Written): The curves showing the variation in the resistance of air spaces with width of air spaces and mean temperature have considerable commercial value in the building trade as well as for manufacturers of refrigerators.

It is significant to note the considerable change in the resistance of an air space with the temperature. The thermal resistance of insulating materials falls off with increasing temperatures. The same is apparently also true of air spaces.

It is apparent from the resistance plotted that an air space equal in width to the thickness of commercial insulating boards does not possess equal insulating value. The resistance of a 7/16-in. air space, for example, at a mean temperature of 60 F is only about one half of the resistance of 7/16-in. rigid insulation occupying the same space. Increasing the width of the air space from 7/16-in. to 1-in. raises its resistance very little so that the results check the recommendation of the Bureau of Standards to the effect that an inch air space be regarded as equivalent to about ¼-in. of insulating material.

In view of the fact that the tables in The Guide are computed upon the assumption that the thermal resistance of an air space is equal to the sum of the two surface resistances, the results of the tests made by the authors would change the values of the tables considerably if adopted. The resistance used in computing the tables is about double the resistance determined in these tests.

In discussing the factors affecting air space coefficients, the authors mention the effect of width of air spaces, mean temperature between surfaces, condition of surface, and the ratio of the area to the thickness of the air space. In buildings an important additional factor would be height of air space in the case of vertical spaces. It would be expected that convection currents in a high air space, as in the walls of a building between studs not blocked by fire stops, would be a larger factor in heat transfer than in the case of the relatively small areas with which the experiments were made. Information on the effect of height would therefore be desirable.

R. H. Heilman (Written): Professor Rowley and Mr. Algren are to be congratulated upon their excellent paper, as the conductance and resistance of air spaces of various widths and at various mean temperatures is of the greatest importance to the engineering profession.

They have shown conclusively that the conductance of an air space increases appreciably as the mean temperature is increased and also that the conductance

decreases as the thickness of the air space increases. Both of these phenomena are as would be expected from a theoretical consideration. The increase in conductance with increase in mean temperature is to be expected as the heat transmitted across the air space by radiation, conduction and convection increases as the temperature increases. The radiation being proportional to the differences of the fourth powers of the absolute temperatures, so that for the same temperature difference between the two faces the radiation will increase as the temperature increases. The conductivity of air increases as the temperature increases, the temperature coefficient per C being 0.0029 as found by Eukens or k for air $= k_o\ (1 + 0.0029\ t)$.

Various investigators have shown that the transfer by convection is approximately proportional to $t^{5/4}$. This value has been confirmed by the writer, and Griffiths has shown that this value also applies to the convection transfer in closed air spaces. Griffiths also shows that the transfer by convection per unit area is also independent of the height of an enclosed air space.

It is to be expected that the conductance would decrease as the thickness of the air space increases as mentioned by Professor Rowley and Mr. Algren, and the writer is of the opinion that the values as given in Fig. 2 are very nearly correct. The results obtained under the direction of the writer at *Mellon Institute* in March, 1928, substantially check the results obtained by Rowley and Algren.

Since there are very little published data on heat transfer across air spaces, a few of the results obtained at *Mellon Institute* may be of interest. In these tests, data were obtained for the conductance across 0.5 in., 0.975 in., and 4.00 in. air spaces. The conductance in most cases was determined for the heat flow upwards, downwards and horizontally.

The tests were made with alundum plate heaters 9 in. in diameter with an air space 8 in. in diameter and the various thicknesses as stated. Heat was transmitted from the alundum plate to the brass cooling plate. Both plates were coated with lampblack so as to be able to determine accurately the heat transmitted by radiation.

The heat flow upward was obtained by using two heating plates and one cooling plate. One heating plate was placed in the bottom of the container in a horizontal position, a layer of insulation was then placed on the heating plate and the second heating plate was placed on top of the insulation. The air space was between the second heating plate and the cooling plate on top. Heat flow downward was prevented, by keeping the two heating plates at the same temperature, then since there was no temperature difference between the two plates, the heat could flow only upward across the air space. The heat flow downward was obtained by reversing the position of the heating plates and the cooling plate.

For the heat flow downward with the 0.975 in. air space, a conductance of 1.390 was obtained with the hot and cold surfaces at 142 F and 53.6 F, respectively, or a mean temperature of 88.4 F. The heat transferred by radiation $= 17.23 \times 10^{-10} \times 0.97\ (602^4 - 513.6^4) = 103$ Btu. The total heat transfer as measured $= 123$ Btu. Since there was no heat lost by convection, the heat flow by pure conduction $= 123 - 103$ or 20 Btu. The conductivity of air then as

obtained from this experiment $= \dfrac{20 \times 0.975}{88.4} = 0.22$ Btu per hour per square foot per degree Fahrenheit per inch thickness.

There is considerable variation in results on the thermal conductivity of air as determined by various investigators, the values at 0 C ranging from 0.000049 to values above 0.00007 in C. G. S. units.

If we take a mean value of approximately 0.00006 we have for a mean temperature of 97.8 F or 36 C a value of $0.00006\ (1 + 0.029 \times 36) \times 2903$ or 0.192 Btu per hour per square foot per degree Fahrenheit per inch thickness, which checks fairly well with the values obtained at *Mellon Institute* when the wide range of values obtained by the various investigators is considered.

The conductance for heat flow upward for the 0.975 in. air space was 1.422 for hot and cold surfaces of 145.5 F and 59.2 F, or a mean temperature of 102.3 F. When we consider the higher mean temperature for the upward flow, it is seen that there is practically no transfer of heat by convection for air spaces 0.975 in. thick in a horizontal position or with the heat flow upward.

The conductance for the heat flow horizontal for the 0.975 in. air space or with the plates in a vertical position was 1.491 for hot and cold surfaces of 137.5 F and 55.6 F, respectively, or a mean temperature of 96.5 F.

When we consider that the conductance should be greater for lampblacked surfaces than for insulation surfaces such as used by Rowley and Algren, and especially since the radiation factor across a 1-in. air space is by far the greatest factor, the conductance of 1.491 as obtained at *Mellon Institute* can be considered to be a satisfactory check on the results of Rowley and Algren who indicate a conductance of 1.35 for 1-in. air space at a mean temperature of 96.5 F as obtained from Table 4.

C. K. SWIFT (WRITTEN): It is rather surprising to note the relatively high conductivity values reported for fibrous insulating boards as shown in Table 1. If my interpretation of these results is correct, the conductivity values are about 8 per cent higher than the generally accepted values for boards of this type.

From the very nature of the test it must be assumed that relatively low plate pressures were used. I would like to know whether the samples were tested, dry, or at current moisture, and what precautions, if any, were observed to prevent condensation of water on the samples during low temperature tests.

The air space conductances as shown in Table 4 seem to be greatly at variance with previously accepted values for still air coefficients. As I understand these values, they include the resistance (or conductance) of two surface films plus that of the intervening air space. A 1-in. space at 60 F mean temperature would, for example, have a conductance of such a space and would give a value of

$$\frac{1}{\dfrac{1}{1.34} - \dfrac{1}{1.34}} = 0.67 \text{ Btu per hour per square foot per degree Fahrenheit. } ?$$

The substitution of Professor Rowley's coefficients for the present accepted values leads to rather serious differences in some cases.

I have estimated the transmission coefficient of a stucco frame wall section as an example. The construction consisted of stucco on expanded metal, fur-

ring, 7/16 in. fibrous insulation, studding, and wood lath and plaster. The transmission coefficient calculated by the ordinary method was 0.169 Btu. To obtain the same overall coefficient using Professor Rowley's values would require slightly over 1 in. of the same type of insulation. In making this calculation, due correction was made for variations in the mean temperatures of the several surfaces, but the same still air coefficients were taken to apply not only to the surfaces of the insulation but to other surfaces as well. While this assumption may not be justified in the absence of definite experimental data it is the best that can be made, and is reasonable in view of the great differences noted in the cases which have been reported.

Where a designer is limited to a definite transmission value for a certain type of wall, and is required to estimate the necessary insulation, the economic importance of accurate surface coefficients becomes strikingly apparent.

I am glad to note that this research is being continued. It is certainly of fundamental importance and should soon eliminate empirical calculations.

W. H. CARRIER: I would like to ask Professor Rowley if he has been able to produce a rational relationship between the increase of conductances of any given air space and the mean absolute temperature? I see that there is a very marked increase of conductance with the mean temperature of the space. Am I correct in that?

PROF. F. B. ROWLEY: Yes. The curves of Fig. 2 extended cross the zero line at from —240 to —243. From this it would appear that if the curves were extended to lower mean temperatures by actual tests, they would not be straight lines. They should gradually curve upward as there seems to be no reason why they should cross at the temperatures of —240. For all practical purposes, the straight line relation is sufficient.

MR. CARRIER: Have you additional figures?

PROFESSOR ROWLEY: We have not worked that out. The process of heat transmission through an air space is rather complex. It is a combination of radiation, convection and conduction.

L. A. HARDING: It might be well to review for a moment the history of air space construction. Thirty years ago air space construction was the standard form of insulation employed in refrigerators, breweries, and cold storage plants. About 26 years ago the Nonpareil Cork Co., ran a very complete series of tests on air space construction, starting with the single air space, and going up to four. I am very glad to see that Professor Rowley has taken up the testing of air space construction with a greater degree of refinement.

Twenty-six years ago, the tests referred to were the only tests that were available to the ventilating and refrigerating engineers. I have been wondering whether Professor Rowley's tests show any very marked differences. Some of the results of these tests I reported to the Society in 1913.

The standard construction was two ⅞-in. boards with paper between; then a ⅞-in. strip; then another layer, etc. The maximum used in any plant in this country at that time so far as I know was the four air space construction.

PERCY NICHOLLS: The history of the measurement of the heat transmission of air spaces goes back further than Mr. Harding intimates. Peclet developed formulae for wider air spaces in 1830 to 1840; he gives no data for narrow

widths but he probably experimented with them because he states that his formulae apply only to spaces over 0.79 in. wide. The authors' curves of Figs. 3 and 4 show limiting values for the conductance with this same width.

In more recent years work has also been done by the Bureau of Standards, the *National Physical Laboratory* of England, and Professor Kruger in Sweden. The two former aimed to separate the conduction and convection factors; full data are not available to compare their results with those of this paper, and moreover the area, or height of the spaces, differs. The *National Physical Laboratory* used apparatus to give spaces 2 and 4 ft high and found the conductance was the same. The Bureau of Standards found a greater conductance for an 8-in. than a 24-in. height. I think the authors might better have worked with one larger than 9 in. square, although the difference for the narrower spaces would not be great.

It is interesting that the authors obtained straight lines in Figs. 1 and 2 for the curve of conductance against mean temperature of the air space; because they do not state to the contrary, one might assume that the temperaure difference between the spaces remained the same with increase of mean temperature. One would expect the conductances to increase with mean temperature, and the fact that they are straight lines would indicate that the temperature difference increased with the mean temperature. The increase of conductance with a mean temperature increase of 100 per cent as shown by Fig. 2, can however be fairly closely accounted for by the increase of the radiation factor from the same cause.

The authors have presented their results in a very convenient form for use in a formulae of built-up walls. Further work to establish values for a greater height should be well worth while, and statements of the temperature differences should be included in the paper.

Prof. J. D. Hoffman: The subject under discussion is one of the most important subjects before the Society. I recall that it was but a few years back when the question of the influence of the house construction upon the heating system came rather prominently before the Society. Since that time we have been considering the house along with the heating system.

The framed house is the chief offender and the framed wall is closely allied with the subject under discussion. For that reason, and because of the fact that probably 75 to 80 per cent of the houses in the country are of this type of construction, I think we can well afford to study the value of insulation as applied to the built-up wall. I congratulate the Society in attempting to solve this problem.

No. 826

AIR INFILTRATION THROUGH VARIOUS TYPES OF BRICK WALL CONSTRUCTION

The results of cooperative research between the University of Wisconsin and the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

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INTRODUCTION

DURING the Fall of 1927, work was begun on a program of cooperative research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the University of Wisconsin. This research was intended to give more definite information relative to air infiltration occurring through plain 13-in. brick walls. The first results of this research were published in a paper entitled Effect of Frame Calking and Storm Windows on Infiltration around and through Windows which was presented at the Semi-Annual meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in June, 1928.

This report is a continuation of the foregoing program and consists of a comparative study of the results obtained with five 13-in. brick walls of different character, with respect to nature of brick, mortar and workmanship.

Description of Test Apparatus

The test apparatus, shown in Fig. 1, is described in detail in the previously mentioned paper. Briefly, it consists of the following: the pressure chamber *A*, and collecting chamber *B*, between which the wall is secured by means of *C* clamps. The method of clamping is more clearly shown in Fig. 2. Air-tight seals are obtained between the two sides of the wall, and chambers *A* and *B*, Fig. 2, by means of a sponge rubber gasket attached to the perimeters of the chamber openings.

Artificial wind pressure is produced by a small motor-driven blower, shown at the extreme left of Fig. 2. This blower is in communication with the pressure chamber through an adjustable damper *E*, by means of which the pressure drop through the wall is controlled. Other control dampers are provided at *D* and on the intake to the blower itself. The pressure difference in chambers *A*

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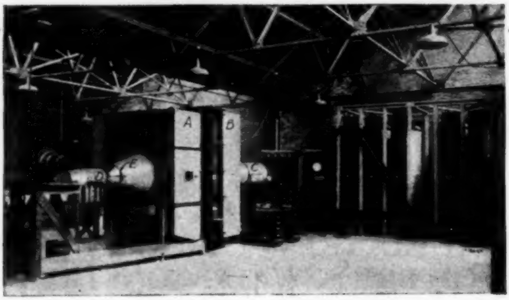

FIG. 1. GENERAL LAYOUT OF TEST EQUIPMENT

and *B*, which is the pressure drop through the wall, is measured with an inclined draft gage, *F*.

The amount of air which filters through the wall is measured by an orifice, mounted on the end of orifice box *C*. The pressure head on the orifice is determined by a Wahlen gage *G*. This pressure head is the difference in pressure between that in the orifice box *C* and in the atmosphere.

Description of Walls

The walls were built into frames constructed of 15-in., 33 lb. steel channels in a manner shown in Fig. 2. The A-shaped frame which appears in the foreground of Fig. 2, together with a single roller jack attached to the opposite end of the wall provides a convenient means of transporting the walls between the test machine and the storage rack shown in Fig. 1.

Two types of brick were used in the construction of these walls—a hard face brick and a more porous type, commonly known as Chicago clay brick. A detailed report of tests conducted on these bricks relative to porosity, compressive strength and other physical characteristics will appear in the final report.

Three of the walls were built up with a cement-lime mortar; the other two with a lime mortar. The walls were built up in such a way as to differentiate

FIG. 2. MACHINE OPEN WITH A BRICK WALL IN PLACE

between good and poor workmanship. Each of these terms is defined as follows:

Lime Mortar. One of lime and three of sand by volume and enough water to make the mixture workable.

Cement-lime mortar. One of cement, one of lime and six of sand by volume, and enough water to make the mixture workable.

Workmanship. Good workmanship is distinguished from poor workmanship only in the manner of using the mortar. In good workmanship, the spaces between the bricks are completely filled with mortar throughout the thickness of the wall, resulting in a wall which is practically free from voids. In poor workmanship, very little mortar is used between the two outside faces of the wall. The outside appearance of these walls is the same. Fig. 3 is intended to show this difference in workmanship. The poorer wall appears in the foreground. The walls were constructed by bricklayers from the Service Department of the University of Wisconsin, and in such a manner as to make their construction comparable to actual building construction practice.

Procedure

Each of the walls was subjected to wind pressure corresponding to a range of wind velocities of about 5 to 30 mph. Except for the initial tests on Wall No. 2, the joint between the steel frame and brick was completely sealed with a plastic calking compound on both sides of the wall, while testing. The effect of calking this joint on one side, and both sides, is shown by the curves on Fig. 4. In determining the net area of wall exposed, allowance was made for the area covered by the calking compound.

After about five months had elapsed from the time of construction, each wall was subjected to eight complete tests over the velocity range previously mentioned. Alternate tests were run in reverse order. These tests were repeated two months later in a similar manner. Thus, the curves on Fig. 5 and Fig. 6 represent the average results from sixteen tests.

Discussion of Results

Fig. 4 shows the effect of calking the mortar joint between the channel frame and the wall. The curves show that calking the second side results in prac-

Table 1. Humidity and Test Results Variations

Wall No.	Av. Humidity Original Tests	Av. Humidity Check Tests	Change in Humidity Expressed in %	Variation in Test Results %
2	74.3	40.5	—45.4	—7.07
3	55.9	63.4	+13.4	—0.27
4	77.2	58.8	—23.8	—0.14
5	65.4	31.7	—51.6	+2.62
6	67.1	36.6	—45.5	—0.24

tically the same additional reduction in leakage as did the calking of the first side.

Fig. 5 shows the results of the tests on the five walls. The infiltration in cubic feet per hour per square foot of wall is plotted against the pressure drop through the wall in inches of water. Two sets of points are shown for each curve; one set from the original tests made five months after construction and

TABLE 2. VALUES FROM THE CURVE FOR WALL 4

Miles per Hour	Cu Ft per Hour per Sq Ft	Cu Ft per Hour per Sq Ft per Mile Wind Velocity
5	0.71	0.14
10	2.36	0.24
15	5.05	0.34
20	8.31	0.42
25	12.08	0.48
30	16.00	0.53

the other from the check tests made two months later. By finding the difference

FIG 3. BRICK WALL UNDER CONSTRUCTION

in values of the original and check test points, it was found that the variation for Walls 3, 4 and 6 was considerably less than one per cent. The check tests on Wall 2 gave 7.1 per cent less infiltration than did the original test and the check tests on Wall 5 gave 2.6 per cent more infiltration than did the original tests. This seems to show that there is no correlation between these results and aging of the walls between the time of the original and check tests. Also, there seems to be no correlation between the humidity at the time of tests and the variation in test results. Table No. 1 shows the humidity and test results variation.

TABLE 3. INFILTRATION IN CUBIC FEET PER HOUR PER SQUARE FOOT OF WALL

Wind Vel.	Wall No. 2	Wall No. 3	Wall No. 4	Wall No. 5	Wall No. 6	1929 GUIDE
5	0.34	0.46	0.71	0.51	1.60	1.80
10	1.30	1.64	2.36	1.83	5.30	4.90
15	2.71	3.45	5.05	3.85	10.35	9.35
20	4.59	5.76	8.31	6.34	16.28	14.50
25	6.85	8.38	12.03	9.22	23.05	20.30
30	9.31	11.30	16.00	12.40	30.80	26.50

Wall No.	Kind of Workmanship	Kind of Mortar	Kind of Brick
2	Good	Cement-lime	Hard
3	Good	Lime	Hard
4	Good	Cement-lime	Porous
5	Poor	Cement-lime	Hard
6	Poor	Lime	Porous

Fig. 6 shows the same results as those plotted in Fig. 5 except that infiltration in cubic feet per hour per square foot of wall is plotted against a uniform scale of velocity in miles per hour instead of against a uniform scale of pressue drop in inches of water. The results given in THE 1929 GUIDE for a 13-in. brick wall have also been plotted in Fig. 6. The curves on this sheet show that infiltration increases rapidly as the wind velocity increases. This is shown in Table 2, for Wall 4.

Table 3 gives the infiltration through the various walls at wind velocities ranging from 5 mph to 30 mph.

The curves show that Wall 6 is considerably poorer than any of the other

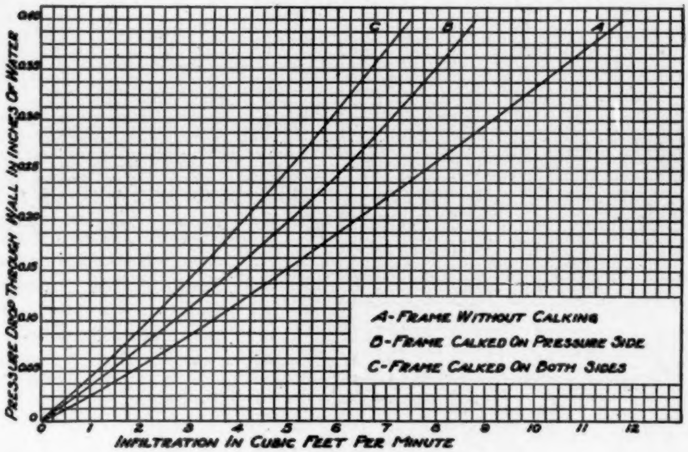

FIG. 4. SHOWS EFFECT OF CALKING THE MORTAR JOINT BETWEEN THE CHANNEL FRAME AND WALL

walls. By improving the workmanship and using cement-lime mortar rather than lime mortar in Wall 4, the infiltration loss is cut to slightly less than 50 per cent. In the case of the hard brick walls, Wall 5, the poorest, allows the passage of 37 per cent as much air as does the poorest wall built of porous brick. The best wall built of hard brick, Wall No. 2, allows the passage of about 70 per cent as much air as the poorest wall built of hard brick, Wall 5. The comparison given here between the poorest and best walls for each type of brick is not strictly true since the poorest of the two hard brick walls had cement-lime mortar as against lime mortar for the poorer porous brick wall.

However, the comparison shows that there is a greater variation in infiltration between the good and the poor walls built of porous brick than between those built of hard brick. This may be due to one or more of several causes. One possible cause is the variation due to chance. Were a similar set of walls to be built the results likely would not check exactly the results of this series of tests because of a variation in materials and workmanship. Also, there is a possible cause in the psychology of good materials. A workman instructed to

do equally poor work on two walls, one built of hard brick and the other of porous brick might unconsciously do better work with the better material, the hard brick. It also seems logical to believe that two walls of good workmanship are more likely to be on a comparable basis from a workmanship standpoint than would two walls of poor workmanship, inasmuch as a completely slushed wall is more easily duplicated.

Since poor workmanship consists mainly in leaving voids between the bricks in the interior of the wall, and since the porous brick is likely to be much less uniform in density it may be that poor workmanship opens up passageways for

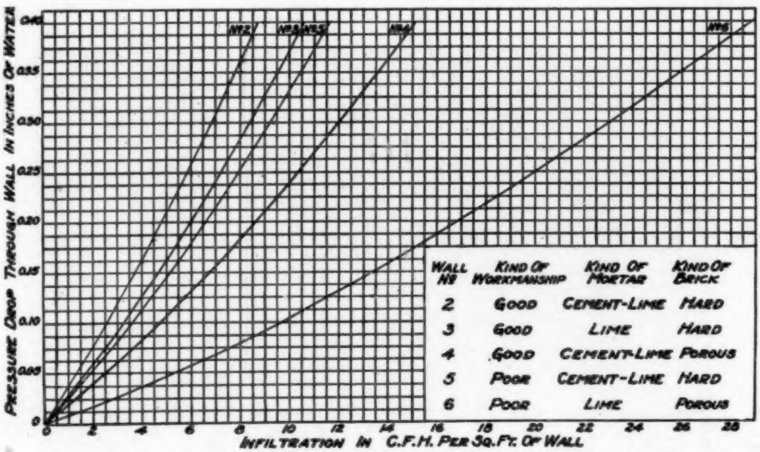

FIG. 5. RESULTS OF TESTS ON THE FIVE WALLS

air through short distances from the face of the porous brick to the voids, and then out on the other face of the wall through a short distance of brick. This explanation requires that the hard brick wall passes most of the total infiltration through the mortar joints.

Another possible cause for this greater variation between the best and poorest of porous brick walls as compared to the best and poorest of hard brick walls is in the effect of the porosity of the brick on the proper setting of the mortar. It is likely that the porous brick draws the water from the mortar before it has time to set and consequently causes an opening of pores and a shrinking away of the mortar from the brick surfaces.

A wall built of hard brick, lime mortar and poor workmanship would have a probable leakage of 4.59 cu ft per hour per square foot at 15 mph. This is equal to the leakage through Wall 5 which was built of hard brick, with cement-lime mortar and poor workmanship plus the difference between the leakage of lime mortar and cement-lime mortar as applied to hard brick Walls 2 and 3 ($3.85 + 0.74 = 4.59$). This then would be a wall built to the same specifications as Wall 6 except for the difference in brick. A comparison shows that this poorest hard brick wall would have a leakage of 44 per cent as great as that through the poorest porous brick wall.

The substitution of cement-lime mortar for the lime mortar in this poorest hard brick wall would reduce the leakage by 0.74 cu ft per hour per square foot of wall at 15 mph or a saving of 16 per cent. The difference in leakage of Walls 5 and 2 gives the comparison of good and poor workmanship for hard brick walls. The saving in using the better mortar is 1.14 cu ft per hour per square foot or a saving of 24.8 per cent. The total reduction in infiltration by using the cement-lime mortar applied with good workmanship in place of lime mortar applied with poor workmanship is 1.88 cu ft per hour per square foot for hard brick walls, or a reduction of 41 per cent.

The same comparison of mortar and workmanship cannot be made individu-

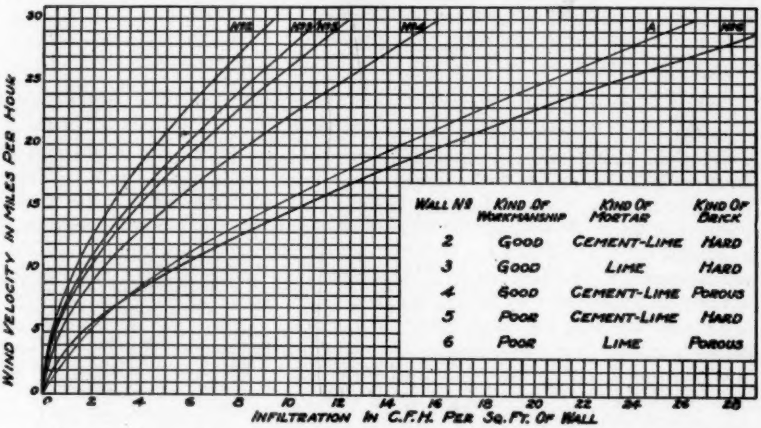

FIG. 6. CHART SHOWING HOW INFILTRATION INCREASES WITH INCREASED WIND VELOCITY

ally for porous brick walls, since only two walls were tested. However, the best porous brick wall on which cement-lime mortar was applied with good workmanship has a leakage of 5.30 cu ft per hour per square foot less than does the poorest porous brick wall, which uses lime mortar applied with poor workmanship, or in per cent the reduction is 51.2 per cent. For hard brick this reduction was 41 per cent. This seems to indicate that it is more important to use the best mortar and workmanship on porous brick walls than on hard brick walls. This is indicated both by the greater saving in cubic feet of infiltration and by the percentage saving. However, the item of costs would also enter in choosing material and workmanship. The hard brick is more expensive and the added cost of cement-lime mortar over lime mortar, and good workmanship over poor workmanship would result in a smaller percentage increase in the wall cost than in the case of porous brick walls.

A hard brick wall with lime mortar applied with poor workmanship would have a leakage of 4.59 cu ft. The porous brick wall with cement-lime mortar and good workmanship has a leakage of 5.05 cu ft. Then the poorest hard brick wall has a leakage 91 per cent as large as that of the best porous brick wall.

Since good workmanship and cement lime were both used on Walls 4 and 2, the difference in leakage, 5.05 — 2.71 = 2.34 cu ft per hour per square foot, seems to indicate the difference between infiltration values of the two bricks. That this difference is not all due to the difference in bricks seems to follow from a consideration of the porosity of the bricks as measured by the water absorption test.

The hard brick had an absorption value of 13.84 and the porous brick a value of 20.69. (These values are to be checked by future independent tests.) If it is assumed that the leakage to air would be in proportion to the absorption to water, the infiltration through the porous brick should be 150 per cent of that through the hard brick. Two equations are then available relating the infiltration through the hard and porous brick.

$$P_b = 1.5\ H_b$$
$$P_b - H_b = 2.34$$

where P_b = leakage per hour per square foot of wall through the porous brick.
and H_b = leakage per hour per square foot of wall through the hard brick.

Solving these simultaneously the infiltration value for hard brick would be 4.68 cu ft per hour per square foot and for porous brick would be 7.02 cu ft per hour per square foot. These values are both greater than the leakage due to the porosity of the brick, to the effect of good workmanship and cement-lime mortar, as shown for Walls 2 and 4. With the assumption made this would mean that the difference for the two bricks, 2.34 cu ft per hour per square foot, was not due entirely to a difference of infiltration through the bricks alone. This points to the conclusion that the same grade of workmanship results in more leakage through cement-lime mortar joints on porous brick than on hard brick. This might be attributed to the drying out effect due to the porous brick absorbing moisture from the mortar during the setting process. This drying out would open the pores of the mortar or cause the mortar to shrink away from the brick surfaces and would prevent a proper bond. This would indicate then that to secure the same infiltration through mortar in a porous brick wall as in a hard brick wall additional care in workmanship would be required to the extent of soaking the bricks before laying.

It seems impossible to know for a wall like Wall 2 just how much of the infiltration is due to workmanship, how much to mortar and how much to the brick itself. Making the guess that 1.0 cu ft of the total of 2.71 cu ft infiltration for Wall 2 is through the brick itself and substituting in equation $P_b = 1.5\ H_b$ the infiltration through the porous brick would be 1.5 cu ft per hour per square foot. The difference then between the infiltration through the two bricks is 0.5 cu ft as compared to 2.34 cu ft as arrived at from a comparison of values for Walls 4 and 2. The difference then, 2.34 — 0.5 = 1.84 cu ft, would be due to the greater drying out effect of the mortar in porous brick construction as compared to that occurring in hard brick construction.

The leakage through an individual brick would be such a small quantity that it seems impossible to find the leakage by testing individual bricks. Also individual bricks would vary in leakage characteristics. To obtain the infiltration through the bricks only, a wall might be built up by using a plastic compound or an asphalt applied hot as a mortar. Sealing off the faces of the mortar joints by painting with asphalt would not give a true determination of the infiltration

through the bricks only. There still would be the chance for air to enter through the faces of the bricks, then enter and travel through the mortar and out through the face of the bricks on the opposite side of the wall. The relation between the water absorption and air infiltration characteristics might be known by testing bricks only.

The tests on walls for infiltration through bricks only would give a means of obtaining the leakage through lime and cement-lime mortars used in walls built of the same bricks. The drying out effect of brick on the mortar joint suggests an interesting study in the building and testing of a hard and a porous brick wall with bricks soaked before laying. For proper comparisons, two additional walls built up of bricks of the same lot and with mortar of the same batch but without the soaking of the bricks should be tested. This method of testing for leakage through the mortar joints combined with the wall tests for infiltration through the bricks only suggested in the previous paragraphs would give a means of obtaining the infiltration through various mortars applied in various ways. Mortar is used mainly for joints between bricks so the tests mentioned would seem to give better determinations than testing mortar by itself, by using in a poured mortar wall, or as a plastered wall with a metal lath as a base.

DISCUSSION

L. B. Lent (Written): This investigation discloses at once that infiltration through brick walls is by no means a constant quantity, but is of widely varying amounts, depending on several factors. The maximum and minimum values obtained for a 15-mile wind velocity are 10.35 and 2.71, respectively; this maximum value thus being 380 per cent of the minimum one—a wide variation, indeed.

It would appear, therefore, that the use of a single average value for air infiltration through brick walls is, at best, a mere approach to accuracy and might result in errors of considerable magnitude.

It is quite apparent from experiments on only these five walls, that the amount of air infiltration through brick walls is governed by several factors, just as is the strength of brick walls. But when we pass from these general observations and attempt to discover the effect of individual factors, the task is not so easy.

Of the three principal variables, brick, mortar and workmanship, the first two can be held constant for most any test series, but workmanship is not easy to control and is perhaps that factor having the greatest influence on results. I am inclined to believe that some, or all, of the discrepancies noted by Professor Larson are due to the variation of this workmanship factor. For this and other reasons it will be interesting to carefully examine these walls, when they are demolished, after the completion of the tests and note the character of the workmanship, especially as it relates to joint filling and adhesion between bricks and mortar, for different grades of workmanship are principally characterized by differences in these two items. In our endeavor to evaluate the effect of each of the three variables, the work of Raisch in Germany may throw some light on the subject. A report of his work is found in the July 28, 1928 issue of *Gesundheits-Ingenieur.* His experiments were on brick walls laid in mortar and on similar walls with all joints sealed with bituminous materials and wax. He also studied the penetrability of mortars alone and of bricks alone.

Some of his findings and conclusions may add useful information and are given briefly herewith:

1. In brick walls which had cured for 2, 3 and 20 months, the longer drying period showed increased air flow through the wall.

2. The amount of air passing through a mortared wall is far in excess of that which one calculates from the data on a single brick; the ratio being approximately 380 to 1.

3. Air penetrability of brick alone is exceedingly small and does not at all represent the flow through the wall, nor has it any specific relation to it.

4. The mortar is far more easily penetrated by air than is the brick. And additions of cement to mortar decrease the penetrability.

It would appear, from the results of both Professor Larsons' and Raisch's investigations, that the character of the mortar joint has much more influence on wall infiltration than the physical qualities of the brick. And in this connection, it is important to know whether the bricks, especially the porous ones, in the University of Wisconsin tests, were wet before laying. Professor Larson's report does not give this information. If the porous bricks were not wet (they usually are in commercial work), they would undoubtedly absorb water from the mortar and so produce a less effective bond. This may well explain some of the discrepancies, as Professor Larson has pointed out.

It is doubtful, in my opinion, if air infiltration through bricks is closely related to their absorption or porosity, as is suggested. Many tests on Chicago bricks (the same porous bricks used in these tests) at the Bureau of Standards show a porosity (measured by the 5-hour boiling method) of between 16 and 17 per cent, instead of the 20 per cent reported by Professor Larson. And this lower percentage would, of course, bring them nearer the value of 13.84 per cent for the hard bricks and so alter this ratio.

A detailed discussion of Professor Larson's ingenious analyses and calculations is not offered herein. Some conclusions of my own are:

1. The apparent discrepancies in results are, I believe, due largely to a variation in the workmanship factor; most difficult to control as Professor Larson points out.

2. The character of the mortar joint has far greater influence on air penetration than the physical properties of the bricks.

3. While cement-lime mortar has less penetrability than lime mortar and the richer cement mortars less than those with less cement, the workmanship factor has more influence than either.

4. That in any future tests, it is desirable, though admittedly difficult, to hold the workmanship factors (both poor and good) as constant as possible, if the effect of other factors is sought.

5. That information published in THE GUIDE be fully informative, pointing out that our knowledge of this subject is in the process of development and that values for air infiltration through brick walls cover a wide range and are not, therefore, fixed or constant.

ARMIN ELMENDORF: I should like to ask if Professor Larson expects to make any air infiltration tests on brick veneer walls.

Sometime ago I had occasion to make a recommendation on insulating a large brick veneer residence in which the framing was of steel. A strong rain drove

against the outside of the brick wall which was only 4 in. thick, sending streams of water through the mortar joints and down on the inside surface. It wasn't necessary to make an infiltration test on a wall of this kind. The many crevices permitting air currents to pass through practically wiped out any insulating value there would be from a perfectly laid brick veneer wall. In computing the heat transmission of a completed wall with brick veneer laid as poorly as in this case, the brick wall would have to be eliminated entirely as a factor contributing insulation.

If the bricks are loosely placed with many openings in mortar, and if this is commonly the case in brick veneer walls, it would seem that heat transmission computations should always omit brick veneer as a factor contributing insulation.

G. L. Larson: I might say in answer that I am not much of a prophet; I don't know what we will get into. Probably as time goes on we will be able to test just such a wall.

H. M. Hart: Does the paper state how long the coat of plaster was allowed to dry?

Professor Larson: About three weeks before the test was made.

E. B. Langenberg: Would it be possible for Professor Larson to take the research work done on chimneys and flues, and get the difference in weight of air between hot inside and cold outside? This would give some indication of the infiltration through the brick. This is a problem that affects every heating man in the United States, and if such data could be put in usable form they would enable the mason contractor to install brick construction for chimneys right and thereby help the heating contractor. This is a problem with which we are continually confronted. In some of the old brick set furnaces, the heat inside has cracked the mortar and separated the brick and we find leakage of dust to the inside. It is difficult to convince the owner that the dust is not coming from the heating plant but rather from the cellar. The only way we can prove it is to put a little pressure on the inside and force smoke through the cracks.

S. R. Lewis: It might be interesting to remember that this work is being done in cooperation with the University of Wisconsin, and the *Common Brick Manufacturers Association* and we believe that the effect of good workmanship probably will be broadcast by the *Common Brick Manufacturers Association.*

E. S. Hallett: The use of lime mortar is mentioned as producing about 25 or 30 per cent increase in infiltration by using such mortar. I am in hopes that the profit to be gained by using lime-cement will be brought out so that we shall have lime-cement instead of lime only.

During the tornado we had in St. Louis about a year ago, several schools were blown down, and all of them were built with lime mortar. One school that was destroyed, had a thousand pupils in it at the time, and two minutes after the tornado struck the school, there was a foot of brick and debris in the yard. Twenty minutes before that the children had all been in the yard. In this school not a child out of a thousand was hurt; yet the school was practically destroyed. The exterior walls all fell out. Other schools in the path of the tornado built at later dates with lime-cement were not damaged although they were subjected to the same intensity of wind, and the same storm passed right over them. The insurance value of lime-cement over lime was the value of a thousand lives in that school. It is difficult, perhaps, to persuade people to use lime-cement mortar

when lime mortar is cheaper, but if the owner knows that he is going to reduce infiltration 25 to 30 per cent and thereby reduce fuel consumption, it ought to be of some value as an inducement.

D. R. BREWSTER: To what extent is this good construction that Professor Larson mentioned (in which the mortar is flushed clear through from the inside to the outside of the joint) actually used in practice in the building of brick walls. The amount of mortar used seems to have a very marked effect upon the amount of infiltration.

PROFESSOR LARSON: We questioned around as much as we could on that particular point and found that the kind of workmanship that we designate as good workmanship is not used very much. Most all brick walls are built like wall No. 6. Look at them from the outside and they look as good as the best, but if you could open them up you would find you have pretty much of a shell as far as mortar is concerned.

L. A. HARDING: That is true the country over and I do not believe you will find many walls constructed today that have anything like slushed full mortar joints in the interior of the wall. I want to corroborate your statement in regard to ordinary lime mortar. That is reflected in the contractor's bid in tearing down old buildings. We allow $6.00 per thousand for brick laid up with lime mortar for salvage; and if they are laid up with good lime-cement mortar, we don't allow anything for salvage.

E. C. EVANS: Were any records kept of the amount of mortar per thousand brick, and will the brick association working with Professor Larson disseminate that kind of information to the trades; because if they don't we will lose the full benefit of these tests?

J. D. CASSELL: This controversy is getting very pertinent with me. Mr. Hallett spoke on the difference between lime mortar and cement. Our program in Philadelphia is to insist upon 10 per cent lime introduced into the cement mortar, but no more. There are preparatory mortars that we wouldn't use because they use more lime than that, and it reduces the strength. Pure cement and sand make the strongest mortar, but it is not so readily laid.

In reference to flushing joints, I will discharge any bricklayer who fails to properly back fill or flush joints, and I oversee probably three hundred bricklayers a week. These men come to us from other places, and we have our superintendents follow them up. If a bricklayer will not flush, that man cannot work for us. It is not a hard thing to do. It is not a union matter. Nearly all of our bricklayers are union men, but the union has nothing to do with that. If you want a job laid up with poor flushing, that is your fault. Insist on a good job, and you will get it.

F. D. MENSING: Regarding chimneys, everybody knows that a chimney is supposed to carry off smoke, and carry it out of the top. A bricklayer or contractor may not realize that it does more than that. When the chimney is all ready, close the bottom, drop a smoke bomb in the top, put a cover over it, go away and watch the result. You will get tight chimneys.

MR. LEWIS: I think it is very interesting that the *Common Brick Manufacturers Association* was the first organization to take up cooperative research with us, and I have every confidence that the information which has been gained by that research will be disseminated by them to the various manufacturers and owners.

No. 827

APPLICATION OF OIL BURNERS TO VARIOUS TYPES OF DOMESTIC HEATING SYSTEMS

By J. H. McIlvaine,[1] Evanston, Ill.

NON-MEMBER

THE designers of oil burners for domestic use have, in general, followed the methods of several types of industrial burners of preparing the liquid fuel for combustion. Certain designs have been combined with others to secure as many advantages as possible and to eliminate in so far as possible the inherent disadvantages. Higher standards of workmanship and new and better materials have been developed to secure quieter combustion, more dependable operation and longer life.

This mechanical development and improvement, however, has only been a small part of the problem of utilizing oil fuel for domestic heating. An industrial burner is usually operated by an experienced man who makes the correct oil and air adjustments as draft and load conditions vary, and the load is usually more or less constant from day to day. The domestic heating load, on the other hand, varies from zero or even a negative demand in warm weather, all the way to full demand in extremely cold weather, and when quick heat is desired.

Besides, the domestic oil burner must be adpated to the four types of domestic heating systems; steam, vapor, hot water and warm air and a fifth, hot water for washing and bathing. Automatically securing and maintaining the desired even temperature in every room while supplying hot water for washing and bathing presents problems of equal importance and demanding even greater engineering ability than the mechanical design of the burner itself.

Besides the various types of mechanical design there are three fundamental methods of operation:

A. Intermittent. *B.* High-Low Continuous Flame. *C.* Graduated Continuous Flame.

A. INTERMITTENT OPERATION

With this method of operation flame is all on or all off. Usually the flame, when on, is adjusted for fuel and air to carry from 20 to 25 per cent in excess of the maximum demand for heat. Thus, during a protracted cold spell, this

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type of burner operates nearly continuously and must be capable of maintaining continuous operation for many hours. On warm days, when no heat is needed, the burner is of course inoperative. Throughout the average heating season such as experienced in the latitudes of Chicago and New York, the burner operates from 1/4 to 1/3 of the time.

(*a*) IGNITION. To start this type of burner automatically, some outside source of heat or flame is required. There are five methods in common use:

1. *Continuous Gas Pilot.* This is a simple method of ignition, but, to insure an adequate flame in case the gas pressure drops, an excessively large flame must be used consuming an expensive amount of gas.

2. *Expanding Gas Pilot.* With this system, the pilot is small during the off period of the burner and is automatically increased when the burner is started. After allowing a sufficient time for ignition, it is then automatically decreased again. The valve controlling the size of the pilot flame is controlled by means of a solenoid or a small motor.

3. *Continuous Electric Spark Ignition.* By means of a transformer the line voltage is stepped up to many thousand volts causing a spark to jump between two electrodes placed so as to ignite the atomized oil. This spark continues as long as the burner is in operation.

4. *Intermittent Spark.* With this system, the spark continues only long enough to ignite the oil. In some burners, the electrodes are then automatically removed from the zone of the atomized oil.

5. *Combination Electric Spark and Gas.* In this system, the electric spark ignites the gas pilot, which in turn ignites the oil.

(*b*) TEMPERATURE CONTROL. There are two methods of maintaining the desired temperature:

1. *By Thermostat.* This device is located at some point in the building which represents as nearly as possible the average temperature of the entire building. It is mounted on the wall about 5 ft from the floor and should never be exposed to the direct heat of the sun, a radiator or an open fire-place. It should not be mounted on a cold outside wall, near hot or cold pipes in the wall, nor exposed to cold or warm drafts from other rooms, halls, doors or windows.

There are two types of thermostats. One type consists of a strip of bi-metallic metal made by welding together two metals having different coefficients of expansion. One end of the strip is rigidly fastened to the frame while the other end bends to one side if affected by a drop in the room temperature, thus making electric contact with a platinum point closing an electric circuit which in turn causes the burner to start. As the room becomes warmer, the bi-metallic strip moves in the opposite direction, makes another contact and causes the burner to stop. The other type of thermostat contains a bellows or diaphragm filled with a volatile liquid. A change in temperature contracts or expands this bellows which actuates the switching mechanism. With either type, the usual temperature differential is 2 deg.

The bi-metallic type is usually used when a low voltage circuit of about 15 volts alternating current is available for the controlling mechanism. The bellows type is usually arranged to actuate a small mercury tube switch. In one end are sealed two conductors in series with the burner motor. A drop in the room temperature causes the bellows to depress the end of the tube carrying the conductors, thus closing the burner motor circuit and starting the burner. An increase in the room temperature expands the bellows thus raising the conductor end of the mercury tube allowing the mercury to flow to the other end, thereby breaking the circuit and stopping the burner.

Both types of thermostats are made with or without clocks. The plain thermostat maintains an even temperature day and night. The clock thermostat automatically reduces the temperature to any desired degree at night and raises it again in the morning. Where this interval is 8 hours or more, a saving in fuel is effected, but, for short periods, the additional heat required to bring the walls, floors, furniture and so forth back to 70 F may offset any saving.

2. *By Boiler Control.* With this system, the pressure in the boiler against a diaphragm or bourdon tube opens or closes the burner circuit by means of a platinum point or mercury tube switch. Diaphragms or tubes of varying sensitivity may be selected to accommodate wide ranges of pressure, and the starting and stopping limits may be adjusted to meet different weather conditions. Of necessity, this type of control is only semi-automatic, but in large buildings, where it is impracticable to locate a thermostat which would properly control the temperatures of many rooms, not connected and occupied by many different tenants, this method of control must be used if automatic operation is desired.

Due to the fact that hot water and warm air heat are used mostly in smaller residences, thermostatic control is effective and control of temperature at the hot water boiler or furnace casing is seldom employed.

Because the intermittent burner in the on position delivers 20 to 25 per cent in excess of the maximum demand for heat, it necessarily raises the pressure in the boiler very rapidly at the start. This is particularly true of steel or copper tube boilers. Consequently the pressure at the boiler rises faster than the heat can reach the thermostat and cause the latter to shut the burner down again. To prevent excessive pressures and blowing off of the safety valve, a boiler control is usually necessary to shut down the burner independently of the thermostat. This also prevents over-storing of heat in the system and resulting wide temperature fluctuations.

With hot water heat there is very little danger due to excessive water temperature, but there may be excessive fluctuation of room temperature unless a boiler temperature limit control is used. As a further refinement with hot water heating, the boiler control switch may be actuated by two temperature bulbs, one placed in the water and one carried through the basement wall to the outside. The water limiting temperatures are then automatically raised in cold weather and lowered in warm weather.

With warm air heat, a spiral coil of bi-metallic metal is inserted into the furnace jacket near the top. This bi-metallic metal is the same as used in a thermostat except that it is capable of withstanding higher temperatures. Its purpose in limiting the temperature in the furnace casing is the same as that of the boiler pressure control and the hot water limit control.

(*c*) Safety Controls. If for any reason the oil fails to ignite, or if while the burner is in the on position, the current goes off and is subsequently turned on again, some means of stopping the burner is necessary.

1. *Stack Safety.* This device consists of a spiral bi-metallic coil similar to the Warm Air Limit Control actuating one or two mercury tube switches depending on the type of circuit used, and serves to maintain the control motor or relay device in the burner on position providing the heat of the products of combustion cause it to reach a certain temperature within a certain specified length of time, 20 seconds or so.

Should the oil fail to ignite, the stack will remain cold, and at the expiration of the allotted time, the burner will be shut off again. The accumulation of soot on the bi-metallic spiral affects the operation of this type of switch, and for this reason, some burners force air through a tube passed through the combustion chamber, the stack safety being actuated by the air blown through this tube.

2. *Radiant Heat Safety.* In this device the radiant heat of the flame takes the place of the heat of the products of combustion to actuate the shut-off device. In one type a black surface absorbs the radiant heat causing it to expand and so shut down the burner. In another type, two bulbs are filled with a gas and connected by a tube at the bottom. One bulb is clear and one is black. The radiant heat of the flame causes the gas to expand in the black bulb forcing a quantity of mercury in the connection tube up into the clear bulb making the necessary electric contacts.

With intermittent operation, all the methods of temperature control using thermostats, boiler controls, electric safety devices, expanding gas pilots, etc., where certain steps must be followed each time there is a call for heat, an accurate timing device is necessary to insure the proper sequence of steps and the correct time intervals. This timing device consists either of a small induction motor which actuates the various gas and oil valves and electric switches, or a system of relays and electric contacts opened and closed by magnets or electrically heated bi-metallic strips.

3. *In the absence* of an electric safety device, some burners use a shut-off in the form of a valve which stops the supply of oil to the burner, or a switch which breaks the burner motor circuit. In the event of a flame failure, a small quantity of oil runs into a bucket which overcomes a counter weight and closes the oil valve or opens the switch. Sometimes the counter weight is pivoted so as to pass dead center and lend its weight to the weight of oil in the bucket serving also to deliver an impact to firmly seat the valve or open the switch.

4. *In some burners* a special thermostatic element serves to shut off the gas in the event that the gas pilot goes out.

5. *Due to rapid evaporation* on starting, particularly with steel and copper tube boilers of small water content, a low water safety is essential. This device consists of a float chamber mounted at the water line between the boiler and the glass gage. A drop in the water line causes the float to depress one end of a mercury tube switch stopping the burner. Even with slow heating boilers of large water content and weight of metal, the low water safety is necessary as the oil-fired boiler is not subjected to the same observation and care as a boiler fired two or three times a day with coal.

(*d*) DOMESTIC HOT WATER CONTROL. If an additional burner is used in a separate hot water heater, the temperature of the water is controlled by the same type of switch as used to limit the temperature of the water in a hot water heating system. With steam or vapor heating, the water at or near the boiling temperature may be used to heat the independent hot water supply by means of a heat transfer unit. This unit consists of a chamber through which the water to be heated is passed. A copper coil carrying the hot boiler water passes through this chamber thus heating the supply water.

With this system, in addition to the regular thermostat and boiler control, a separate water temperature control switch is placed in the circuit in such a way that if the room thermostat is not calling for heat, and as a result the temperature of the boiler water falls so that insufficient heat is delivered to the heat transfer unit, the burner is automatically started. When the temperature of the boiler reaches 190 or 200 F, if the room thermostat is still not calling for heat, the burner is shut off before any heat is delivered up into the building.

Some boilers contain a built-in heat transfer unit in the form of a tube immersed in the boiler water, usually just below the water line.

(*e*) ADVANTAGES OF INTERMITTENT OPERATION.

1. Intermittent burners are fully automatic. No attention is required other than keeping the storage tank filled and periodically oiling and cleaning the mechanism.

2. A fairly low-priced fuel may be used, as the electrical load of the atomizing mechanism may be considerable without an excessive cost of electricity because the intermittent burner only operates on the average one-fourth to one-third of the time during the heating season.

3. No fuel is wasted during warm weather in the spring and fall.

4. Long life because the wearing parts are in operation only one-fourth to one-third of the time.

(*f*) DISADVANTAGES.

1. *Complicated ignition devices.* If close temperature regulation is desired, the ignition cycle may be repeated as often as 12 to 24 times a day or 2500 to 5000 times during the heating season.

2. *Rapid expansion* of gases on ignition. When the burner has been inoperative for a considerable length of time, the stack draft is insufficient on starting to carry away the products of combustion unless some means of effecting a gradual start is employed. If ignition is delayed until the boiler, flue and stack are filled with a combustible mixture, sudden ignition may cause such a rapid expansion of gases as to damage the boiler.

3. *If an excessive number* of starts and stops is to be avoided, the intermittent heat will cause overrunning and subsequent wide fluctuations of room temperature.

4. *Heat is lost* during average winter weather by excessive stack temperature in the on position and subsequent cooling during the off position. Heat is also lost through incomplete combustion when starting from a cold boiler. Close temperature regulation with frequent starts and short running periods increases the heat loss.

5. *Alternate expansion* and contraction. This is particularly true of warm air furnaces where some means of absorbing heat during the on position and giving it up again during the off position must be employed to protect the furnace.

B. HIGH-LOW CONTINUOUS FLAME

As the name would indicate, this type of burner operates on a low flame when there is little or no demand for heat and a high flame when heat is required. The low flame serves as a pilot light and must be sufficiently small so as to avoid waste of fuel and discomfort during unseasonable warm days in the spring and fall. As with the intermittent type of burner, the high flame should deliver from 20 to 25 per cent in excess of the maximum demand for heat.

Oil and air supply controls are usually interlocking and must be adjusted for both operating positions. If mechanical draft is used, the burner motor and other moving parts must be properly designed to operate continuously throughout the heating season.

(*a*) IGNITION is manual by means of a gas or oil torch and no automatic ignition devices are required. If the oil is prepared for combustion by vaporization, sufficient time is required to preheat the vaporizing plate or chamber. If the oil is atomized, ignition may be without preheating.

(*b*) TEMPERATURE CONTROL.

1. *Thermostatic.* As with the intermittent type of burner, the room temperature may be controlled by a thermostat, but instead of starting and stopping the burner, the flame is turned high or low. If full automatic temperature control is desired, the flame is operated throughout its complete range each time the demand for heat changes. This method is usually employed in steam or vapor systems if the full flame is required to vent the system. In hot water and warm air systems, the operation is sometimes made semi-automatic, the range and limits of the size of the flame being adjusted in accordance with weather conditions.

2. *Where the boiler* pressure or water temperature is used to control the flame, an electric regulating device may be used, or if the burner controls offer very little friction, they may be operated by means of the coal draught regulator providing snap action from high to low and back.

(*c*) WITH THERMOSTATIC CONTROL a boiler limit control is required as well as a safety shut-off in the event of flame failure. A low-water shut-off is also essential.

(*d*) WATER for domestic use may be heated by high-low operation in a manner similar to that of the intermittently operated burners.

(*e*) ADVANTAGES OF HIGH-LOW OPERATION.

1. Complicated ignition devices are eliminated.

2. The combustion chamber, the boiler, and the stack are always warm and the draught is capable of carrying away the expansion of gases when the flame is increased.

3. The boiler or furnace being warm, heat is more quickly generated and delivered to the system.

4. Inefficient combustion in a cold combustion chamber is avoided.

5. With semi-automatic thermostatic control, wide fluctuations of temperature are avoided.

6. Failure of ignition and over-rapid expansion of gases due to delayed ignition are avoided.

(*f*) Disadvantages

1. Manual starting is required.

2. In smaller installations the electrical load of the atomizing mechanism must be low to avoid excessive relative current cost; consequently only higher priced fuels can be used.

3. With mechanical draft the motor and mechanism operate continuously and wear excessively unless oversized and made of the most durable materials.

4. When no heat is required, fuel is wasted to maintain the low flame.

5. If full-high flame is used in average weather, except where steam or vapor conditions require the maximum flame, heat is wasted.

6. Oil and air must be adjusted for two rates of combustion.

C. GRADUATED CONTINUOUS FLAME

In this type of burner, the operation is continuous as with the high-low type with high and low limits of combustion rate to satisfy the greatest and least demands for heat. However, instead of only two points, the burner is arranged to operate at several intermediate steps so that the rate of combustion is proportioned to the demand for heat. With manual operation, the oil and air may be separately adjusted to any required rate of combustion. But for automatic operation, the oil and air controls must be interlocking and must be capable of permanent separate adjustment at the several intermediate steps. With mechanical draft, the burner motor and other moving parts must be designed for continuous service.

(*a*) Ignition is manual and similar to that of the high-low type.

(*b*) Temperature Control.

1. *Thermostatic.* To secure the full advantages of the graduated continuous type, the ordinary high-low type of heat regulator is inadequate. The motor-driven regulator actuating the oil and air controls must be capable of stopping at the several intermediate positions of the controls. Likewise, between its high and low contact points, the thermostat must make intermediate contacts corresponding to the several intermediate positions of the motor-driven regulator at the burner.

Thus a very slight increase or decrease in the room temperature causes the thermostat to switch the burner regulator down or up one step to balance the heat loss of the building. The burner is only operated continuously on the extreme high and low positions when the demand for heat reaches a maximum or minimum.

2. *Boiler Control.* As with the thermostatic control, the graduated type may be arranged to operate at several intermediate positions as determined by the pressure in the boiler or temperature of the water. If the moving parts of the control apparatus are light and free from friction they may be actuated gradually by means of the coal damper regulator.

With graduated thermostatic control, the increase in heat is not so severe as with the intermittent and high-low types, but a boiler limit control is necessary in the event that a window is left open so as to cause the thermostat to call for heat and cause the boiler pressure to become excessive. The same holds true for hot water and warm air systems.

3. *Manual Control.* Where the cost of automatic temperature controls is a factor, the graduated continuous type may be operated manually at the burner or by means of chains or the like brought to a convenient place upstairs. Except for sudden changes of temperature or when it is desired to leave the building for a considerable length of time, manual control is satisfactory. Under certain conditions where an operator is in more or less constant attendance, manual control may be preferable.

(*c*) SAFETY CONTROLS.

1. *Shut-Off.* With manual control and constant attention on the part of an operator, no safety shut-off is needed, but under practically all conditions and certainly with automatic control, a safety shut-off is required in the event of flame failure.

2. A low-water shut-off is also advisable even with manual control.

(*d*) DOMESTIC HOT WATER may be heated in a separate heater or by means of a heat transfer unit with controls similar to those used with intermittent and high-low burners.

(*e*) ADVANTAGES OF GRADUATED CONTINUOUS OPERATION.

1. Complicated ignition devices are eliminated.

2. The combustion chamber, boiler and stack temperatures are gradually increased or decreased and soon adjust themselves to a change in the rate of combustion avoiding periods of inefficient operation.

3. The boiler or furnace responds quickly to a change in the demand for heat.

4. Inefficient combustion due to starting in a cold combustion chamber or suddenly increasing the rate of combustion are avoided.

5. A constant supply of heat is delivered to the heating system insuring continuous circulation and avoiding temperature fluctuations.

6. Heat is not wasted by alternately forcing the boiler or furnace and cooling it off again.

7. Boiler and furnace are not subjected to alternate expansion and contraction.

(*f*) DISADVANTAGES OF GRADUATED CONTINUOUS OPERATION.

1. Manual starting is required.

2. In smaller heating plants the electrical load of the atomizing mechanism must be low to avoid excessive relative current cost; consequently only the higher priced fuels can be used.

3. With mechanical draft, the motor and moving parts operate continuously and must be made oversized and of extra durable materials to avoid excessive wear.

4. When no heat is required, fuel is wasted to maintain the low flame.

5. Oil and air must be adjusted for each operating step.

The proper application of the various types of oil burners to the many existing types of boilers and furnaces and the selection of domestic boilers and furnaces to give the most satisfactory and efficient results with oil fuel are such broad subjects that no attempt has been made to cover them in this paper.

The *American Oil Burner Association* as well as the individual oil burner manufacturers are conducting extensive research programs to determine as far as possible the combinations which will effect the best results. It is to be hoped that the improvement of oil burners and controls will be met by a corresponding development in boilers and furnaces designed especially for oil fuel and that a closer cooperation between these two branches of the heating industry will result in securing the full advantages of automatic oil heat.

DISCUSSION

E. C. EVANS: Nothing is touched on in this paper about confining or absorbing sound energy. Has anything been done to cut down the sound of an oil burner? I am speaking of residential burners.

J. H. MCILVAINE: The design has been improved tremendously in the last three or four years; principally, the velocity of the air supporting combustion has been reduced. With the industrial type of burner, using a heavy oil a high

pressure is required to break up the oil. A lower air pressure may be used with the lighter oils thereby resulting in a so-called softer flame which reduces the noise of combustion to a great extent.

MR. EVANS: With the control now possible with oil burners, it seems that wonderful progress has been made. There is no question about the advantages of doing away with the coal heater (of which I am a user) but the oil burners I have heard in the residences of my friends, in the still of the night would be extremely objectionable.

It should be possible to lead the sound energy out of the combustion chamber. You frequently do that in ventilation work. It helps with our troubles with organ blowers, etc.; it is a well-known fact to our entire body that sound is energy and can be made to travel the way you want it like educated arrows.

That point should be emphasized because I think the progress of the oil burner industry has been and will continue to be seriously impeded until they lick the very objectionable feature of sound.

H. M. HART: This is an excellent paper and one by which the heating contractor, as well as engineer, will be greatly benefited, and I think we are indebted to Mr. McIlvaine for the very comprehensive manner in which these different devices have been explained. Nothing is said about the efficiency under different rates of combustion. I am particularly interested in that in reference to the graduated flame. At what point of combustion does the greatest efficiency prevail; is it the same degree of efficiency all through the different rates of combustion, or is there a difference? If so, how much difference?

H. R. LINN: I have been a very careful observer of the oil burner in domestic service for about ten or twelve years.

Mr. Evans raises a point that has been ironed out and done away with. You can get a dozen burners today that make so little noise that you can scarcely hear them when you are upstairs. I was in Lake Forest a month or so ago, dining with a family, and the man remarked that he had an oil burner in his house. I asked him if it made any noise, and he replied that he did not know. "Let's see! (Listening) It is not running. (Still listening) Yes it is, too."

That is not exaggerated. The subject of noise has been promulgated by those opposed to oil burning, and should be given no credence as there is nothing to it.

As far as the rates of combustion are concerned, we have run a lot of tests on oil burners. There is little difference in efficiency at different rates of combustion. If a man asks me what burner he should buy, there are three things I tell him to look out for. *First,* how well serviced the burner is in his territory. You wouldn't buy an automobile that was made in Melbourne, Australia, and had to send to Melbourne to get some one to fix it. The same thing applies to an oil burner.

Second: How quietly it runs. There are some burners that do not run quietly, but so many do run quietly that he should be careful that he is getting one that is in the quiet class.

The *third* is Efficiency. The efficiency side of the burner will be forgotten long after you remember having had to send to Melbourne, Australia, for a repair man to come and repair it.

I think this paper is wonderful. I think this Society has paid entirely too little attention to this new industry. Our Society has spent literally thousands and thousands of dollars in research that has benefitted one industry largely, and now we are not doing anything to help the Oil Burner Industry. I don't know why we should not start some research work in oil burners. It certainly is one of the coming modes of heating homes, and we are all interested today in making our homes more comfortable. We pay a lot more for things today than we did ten, fifteen, twenty or twenty-five years ago, and yet for some reason or other we look askance at the oil burner.

Mr. McIlvaine touched very lightly on one subject that I wish he had gone a little more thoroughly into; that was the adapting of the oil burner to the existing heating systems. Some burner manufacturers have been more successful in that than others.

I am inclined to think there is a lot that can be done on that. I thought some three or four years ago I had struck at a panacea of all troubles of getting the right burner in the right boiler by classifying the types of flame, and I classified those flames in four ways: the pot type flame, the gun type shaped flame, and the flame which whirls around horizontally and another flame which is largely disappearing from the domestic field, that was the straight shot flame. Then they commenced getting combinations of these flames, and I lost out entirely.

In closing I want to suggest that we give consideration to doing a little research work for this industry which will benefit so many thousands of homes.

B. K. EATON: I have enjoyed Mr. McIlvaine's paper very much indeed. I do not happen to be associated with Mr. McIlvaine in any other way than as a very friendly competitor but I do recognize and appreciate the value of the work that he has done preliminary to this paper, and the very fair manner in which he has presented the merits and demerits, if there be any, of the various types of burners.

There is one thing that is of distinct interest to me. That is the cooperation that this organization can give. In the design of domestic heating systems for oil burners there are two vital defects that we meet in a great many systems. The biggest defect is the chimney. A great deal of work has been done by this organization and by some of the large boiler manufacturers, but the fact remains that nine out of ten complaints are traceable to the chimney. We run into some peculiar situations.

I mentioned one a few years ago before the *Oil Burner Association* and a heating engineer present came to me later and said "I enjoyed the illustration, but I doubt it," yet the illustration is true. This is it:

An architect on the West Coast was designing a six-story building to be used as a dry goods store (the Rankin Dry Goods Co., Santa Anna, Calif.) and he called upon an engineer for a specification. The engineer specified a 12 x 12 flue for that particular building and the particular fuel in use. Another architect, for the Friends' Church at Whittier, Calif., called upon this same engineer to make a specification, and his specification for that particular job was a 12 x 12 flue, the heating plants being practically identical in load and boiler. When these two jobs were built neither would work. At the Friends' Church at Whittier the 12 x 12 flue had been increased by the well-meaning architect to 24 x 24, thereby, as he said, doubling the stack. You know what that means.

In the other job the stack was built in a party wall, and when they began to figure dimensions they decided on a flue 20 in. long and 4 in. wide, since they would have the same perimeter as the 12 x 12.

Both troubles were eliminated through a correction of the chimney.

There are today a number of new style jacketed boilers made by different boiler companies. We find that there is a tendency on the part of the steam heating contractor when he installs such boilers to depend upon the patent insulated metal jacket to entirely diminish the loss of heat.

I installed one of these boilers in my own home which I built this last summer. Upon taking draft readings I found that I had 0.15 in. draft in the base of the stack. I had 0.05 in. draft in the burner. Where did the other 0.10 go? A beautiful looking job—but too great a draft loss.

We took the jacket off the boiler and with a candle test found that there was not a single inch between the sections where the flame did not pull in. Ten pounds of boiler putty, a replacement of the jacket, and the draft was 0.09 in the burner firepot instead of 0.05.

There has been a great amount of practical research work done in the field by the responsible oil burner companies. I think some of this lackadaisical attitude that Mr. Linn has spoken of has been due to the fact that there have been thousands of oil burner manufacturers, whereas there is a small group of high grade, progressive, intelligent oil burner manufacturers, each of whom maintains a laboratory to study these very problems—of which problems, noise was one of the first.

Now, take this matter of the complication of ignition. It has been eliminated in the case of the gas type burner through the use of the old fashioned gas pressure regulating valve, which maintains pressure at the burner irrespective of pressure fluctuations in the gas line.

One of the large regulator companies has developed a robot for controlling electric ignition which is as wonderful to us today as was the thermostat of 40 years ago.

I do believe, if this organization, of which I am proud to be a member, will follow Mr. Linn's suggestion and recognize the oil burner industry as a tremendous industry, forgetting the curbstone oil burner manufacturer just as they forget the curbstone plumber and cooperating with the leading oil burner manufacturers, coordinating, if you please, the work of their laboratories (and I do not speak idly, there are at least half a dozen excellent laboratories in existence) there is going to be a better feeling between the oil burner leaders and members of this organization. There is also going to be a greater advance in the happiness, satisfaction and success attained from the use of heating plants in general, as well as those which use oil burning equipment.

MR. HART: The gentleman who just spoke brought up some of the weaknesses of the heating system for oil burner application, and he overlooked one that I think is quite important, but which the oil burner manufacturer seems to require of the heating design; that is this: Mr. McIlvaine has given us the information that there are cures, but I don't think that the oil burner industry is as familiar with this paper as they should be. I am speaking in reference to control. My experience has been with residences of the larger type and

through lack of cooperation between the oil burner organization and the heating contractor, we have had nothing to say about the oil burner that went in or the method of control, and it has usually been of the ordinary type of oil burner, wherein the flame was controlled from one thermostat placed in one room of about a twenty-room house.

Anyone who has had the experience of trying to get results from such an installation knows that it cannot be done. The oil burner man expects a heating system to deliver instantaneously the same amount of heat from a boiler to every radiator in the house.

The heating contractors have had serious difficulties with their systems with such installations, and I know of a number of heating contractors who decline to be responsible for their guarantee for results when an oil burner is installed on their system because their system is not designed for operation with a fire such as the oil burner will give, and that is a flame that is all on or all off.

The only way we have been able to secure satisfactory results is to ask for continuous operation. We have asked several burner manufacturers if it were not possible to change their control so as to make the limiting device on the boiler the master control and the thermostat the secondary control, and in every instance I have been told that it could not be done. The same thing applies to a vapor system with a big house, the thermostat being located downstairs where the rooms are not allowed to cool at night time and the bed rooms are allowed to cool; the starting load in the morning is very light to bring up the temperature in the room in which the thermostat is located; it comes up quickly; in fact, it comes up and shuts off the oil burner before steam ever gets to the far radiators.

Those are the difficulties that we have in large residence heating. It does not apply to the small residence—that is simple—the radiators and rooms are not so far scattered, but I think that closer cooperation between the oil burner man, heating engineer and the heating contractor would help to a great extent in ironing out these difficulties.

One other experience we have found is this: that the oil burner manufacturer in trying to play safe sometimes has based the size of his burner on the rating of the boiler. Taking the ordinary heating boiler, as rated today and putting in a flame that will allow for 25 per cent in excess of the rating of the boiler makes a very distorted system and adds to the difficulties of control.

E. B. LANGENBERG: Speaking of automatic control, I installed a job not long ago, and I told the owner that there was only one other automatic control I could put on, and that would be a control on the oil tank; when the oil reached about 4 in. from the bottom, a bell would start ringing and could not be stopped until the oil in the tank was replenished.

It ran along for two weeks and the owner called me up and said, "What will it cost to put in that automatic control? I put 2500 gal in my tank, the tank sprung a leak and the oil is out in the dirt, and I have to have some oil."

The public will pay for convenience. It is coming in automatic stokers; it is coming in oil. After we get the oil, we are going to get the gas. They have their troubles with condensation in the flue. That is one thing I want to talk about, condensation in the flue with oil burners. The on and off periods get so

far apart that your boiler chills down, and you get gluey substances in strings 6 or 8 ft long. We had occasion to wreck a flue and discovered that condition in this flue.

We have worked out a solution to offset this condition where you use a refractory material in the firepot. There is a certain amount of latent heat held in that refractory material for a definite length of time, and it has a tendency to keep the flue a little warmer than where you impinge the flame on a cold surface, such as a cast-iron boiler where the chilling effect has a tendency to create soot and smoke, and the other things that make an oil burner work unsatisfactorily.

We had a talk with all of the oil burner men in St. Louis and said "You must quit passing the buck to us in the heating business. We are not going to pass the buck any more to the oil burner man. Will you do so?"

They said "Yes"; they had had their load of trouble. When we come to a trouble job, the two of us get together. My man checks according to the Standard Code. We figure the heat losses to be sure they have been taken care of; then we get together with the oil burner man and say, "What is the most efficient rate of combustion you can put in this furnace to take care of the heat losses?" And he sits down and figures, and a lot of these fellows do not know about oil burners except the sales end, and we insist that they find what the Btu output is.

We took a number of their burners and put them through a test, and discovered we had outputs that varied greatly, and the difference in cost of the burners was about $250. We have ironed out our troubles as far as the consumer is concerned, by working together. It is the very thing Mr. Hart has mentioned and the best we can do.

Another thing we found was the difference in control between gravity warm air heating and fan systems using the Sirocco type fan to develop a pressure on the system, especially in the larger houses over twelve rooms; we find that we cannot turn the fan on for 6½ min after the oil burner comes on. There seems to be a period of between 6 and 7 min no matter how hard you run your burner.

Consequently, we developed a relay so the thermostat that controls the system upstairs turns on the oil burner; as soon as the temperature in the canopy reaches between 200 and 400 F the fan turns on. As soon as the thermostat upstairs reacts both fan and burner are turned off at the same time.

By this method we have been able to maintain a temperature throughout the house that does not vary more than practically 1½ deg in temperature in every room in the house. In windy weather with wind velocities in excess of 20 mph, that varies. It varies also in water or steam jobs.

Flue temperatures vary considerably in different types. The opinion has been expressed that for burning any particular type of fuel, it is necessary to have apparatus built for that particular fuel to obtain the highest efficiency. Many oil burners have been put into operation with small combustion chambers. You cannot get proper combustion, and that has to be corrected.

A number of manufacturers have already built apparatus especially adapted to oil burning. I have one in mind we are working on at the present time. It has developed practically 89 per cent efficiency and a flue-gas temperature of practically 300 F. About six other companies are working on the same problem, and no doubt they will come out in the next year with new designs.

One thing we have suggested to the oil burner people in St. Louis is that when an oil burner is installed in apparatus already in use, the apparatus be torn down and cemented, not just plastered up with cement in the cracks. You have to unmount the whole thing and put fresh cement in to get a key set; otherwise you are going to have trouble. We have had several damage suits where the house smoked up, and part of it was caused by wrong combustion in the chamber, and the other the apparatus was not cleaned out.

We are recommending that in all changes from soft coal to gas, oil or hard coal, the plants be thoroughly cemented and checked over at the time. By working the two industries together we are beginning to get results worth while.

As far as noise is concerned, we are not bothered much with that any more; they have found they can build boxes in front of the noisy ones and eliminate the noise. People are getting used to ice and sewing machines and if they have fifteen kiddies in the house they do not worry about noise anyway.

LAURA A. CAUBLE: I am chairman of the *National Conference Board of Sanitation,* and I am tremendously interested in the discussion. It would seem to me, from the ease with which this discussion has gone on, nearly all your problems are solved.

There is a problem, however, that is not solved, and we want you to solve it. We are dead tired of the infernal smoke of heating plants.

Now then, that must be cleaned up because we insist on having clean air for cities. We insist on it for any kind of heating apparatus. We have been making a study for three years of the loss of sunlight in the City of New York. This has been done in cooperation with the Public Health Service of the United States government, and we find that there is a constant loss of from 35 to 89 per cent of the available sunlight in New York City. You who come from there know that it is not the dirtiest city in the world, although it has become unnecessarily dirty in the last 8 or 10 years, due to the soft coal and oil which is now being burned in the city ignorantly.

We are saying to the Federation of Women's Clubs and to all organized business men who care a whit about what is going on, that we will not sign any more contracts for building heating apparatus in our houses unless there shall be an absolute guarantee of smokelessness, and we are asking that of busimen generally.

We are a democracy. In the early days we did not like a tax on tea, not a bit; we went to war for a release of taxation without representation, and yet we are standing an invisible tax of from $16 to $20 per capita in all the municipalities in New York; that tax amounts to $96,000,000 a year, if we base it on the estimate of $16 per capita, and the loss is in the dirt and filth that comes into our homes.

Now, this is just an economic loss, and there is not a question in that about the loss of time through illness through respiratory sicknesses or what not.

The time has come for a showdown. We do not tolerate a dirty old oil lamp in our homes. In the days when we did have oil lamps, there was some one person who usually got the job of cleaning, some one who had the patience to insure a clean flame as long as the family wanted to use the light.

I know as a housekeeper and a trained woman, that something of that same

sort of cleanliness of method has to be introduced into the burning of these dirty, smoky fuels. I believe it is perfectly possible for the engineers of this country to solve that problem, and do it pretty quick. It would seem from the ease of this meeting that the solution of this problem is almost in your hands, if you would forget some of these other things, and go at it, because nearly everything is done.

Now let us have systems of burning fuels of all kinds that relieve us of the smoky output, that relieve us of the cinders, that clean out the gases, so there is a measure of safety to the citizen who is perfectly defenseless.

I have been asked to come here by your own organization. I am happy in the privilege of coming. There is no conflict between the householders and business men of the country, and the people who do this work. We say, here is your problem; please solve it for us; but from now on there will be a constant effort for clean air for cities and we will not sign any contracts for heating apparatus which do not guarantee smokelessness, and we are not going to rent our homes in apartment houses which are smoky and dirty, and which are adjoining those which are.

That is one way of getting at the problem, is it not, from the standpoint of the consumer?

Mr. Langenberg: I would like to state that I was connected with the Smoke Abatement League of the City of St. Louis for about three years. We raised $250,000 to educate the public. We instituted some research work and then tested four types of warm air furnaces, and two types of boilers. We did find a way by which we could eliminate between 70 and 75 per cent of the smoke and that was by proper firing. The response we met from the public was gratifying and a great deal has been done. It does, however, require an aroused public opinion to carry such a campaign to a successful conclusion. Even strong legislation is required.

They are doing it in Chicago in their politics. But the thing is here. I am glad the women have taken hold of that problem. If that action is taken that you have stated, "that we will refuse to buy a house or live in a house that has a piece of apparatus that smokes," we will whip this thing quickly, and if the women of this country are interested and will do this, I know this organization will cooperate. We are delighted to do so, knowing the inefficiency of most of these plants is caused by incorrect combustion, and unless we get correct combustion we do not get 100 per cent out of our plants. Our living depends upon the good will of the public.

Mr. Eaton: There are among the leaders of the oil burner industry many different burners that do not operate the way those nondescript curbstone burners perform. There is not a single burner sold in Chicago today, at least by members of the Chicago Association, in which there is any necessity for any owner to go near that burner and do any cleaning to eliminate smoke.

Answering Mr. Hart's remarks in regard to big homes, there is one company at least in Chicago that has about four thousand burners in use there, of which a fair percentage are in houses of more than fifteen rooms. They have faced the same problem, in a measure, that you speak of, but not to the extent that you mention. However, we have run into that arbitrary attitude of some steamfitters, so different from the attitude of the warm air man, the attitude of the

latter being "Let's cooperate"; the attitude of a few steamfitters being "We will not guarantee our system if you put an oil burner in." All we want the heating contractor to do is to say "If you will deliver 180 F at the boiler, or 2 lb, or so many ounces of vapor, we have the radiation there to adequately heat every room." The oil burner dealer will guarantee to produce those conditions in the boiler.

If you have a hot water job and you cannot get uniformity of temperature your radiation is not properly proportioned. Then you can choke the radiators in certain rooms to get that uniformity. If it is a vapor system you can throttle the flow of steam into those particular radiators where you get too much heat. Where the steamfitter has correctly determined the heat losses and has figured the radiation on a proportional basis these troubles will not materialize. Indeed, you will find that taking the heating season as a whole, the automatic oil burner will maintain far evener temperatures than coal could ever hope to accomplish.

Thornton Lewis: Our Society is very sympathetic toward making clean cities, not only from the standpoint of combustion, but from many other standpoints that tend to make dirty cities. You will find here a very hearty and sympathetic response.

Mr. McIlvaine: In regard to the efficiencies at the various rates of com-

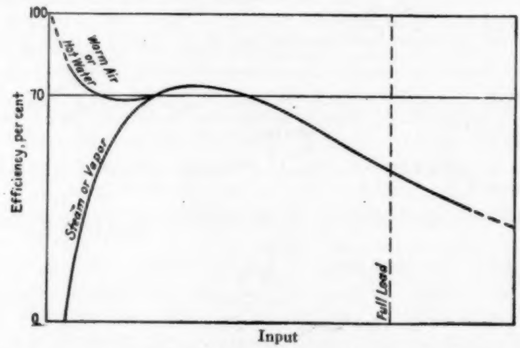

Fig. 1. Oil Burner Installation, Typical Overall Efficiency Curves for Warm Air or Hot Water and Steam or Vapor Heating Systems

bustion, at the close of my paper, I said something about the proper application of the various types of burners to the existing types of boilers and furnaces. Mr. Linn's remarks are on this subject, which is one that would take such a lengthy time for preparation that I did not include it in this paper. It would require a paper three times as long as this one to cover the discussion of those efficiencies.

With a steam or vapor job, the efficiency will be zero when burning oil below a certain rate because you get no steam up into the system. On the other hand, with the warm air or hot water job, the efficiency approaches 100 per cent as you decrease the amount of heat delivered to the system, because, theoretically,

if you could cool down the gases to the initial temperature you would get all the heat out of them.

If you force an infinitely large amount of heat through the boiler, you approach an efficiency of zero so that, some place your efficiency would be, we will say, 70 per cent, probably at around one-third full load, depending upon the size of the boiler passages. If they are small, you get a turbulence at a lower rate of combustion. If they are large, the critical point is moved farther on. With the hot water and steam systems you would get curves something like those shown on Fig. 1. This critical point depends upon the cross section of the boiler flues, and their length.

MR. HART: I am not speaking of the efficiency of the heating apparatus, only of the oil burner.

MR. MCILVAINE: I thought you meant over-all efficiency. You mean combustion efficiency. The intermittent type of burner is usually designed to give

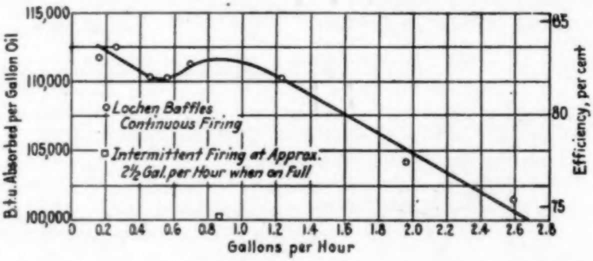

FIG. 2. EFFICIENCY CURVE OF A COMMON TYPE OF OIL BURNER

its maximum combustion efficiency at the rated load. There are different sizes of burners for different sizes of boilers.

The continuous or high-low burner is bound to have a lower combustion efficiency at the lower rate of combustion because in order to maintain a bright clean flame, you have to increase the proportion of air supplied for the combustion. A typical oil burner efficiency curve is shown on Fig. 2.

MR. HART: I asked at what rate of output you got the highest combustion efficiency over the range. You have a range of combustion of, say, from 20 per cent to 100 per cent in pounds of oil; at what rate of that percentage would you get the highest efficiency, and what would the range be?

MR. MCILVAINE: That is a hard question to answer. The efficiency varies so much with different types of apparatus. I would be glad to show you some figures if I had time.

I think the result of all the discussion is that we hope next year we shall have the opportunity of presenting another paper that will take up some of the points brought up, and I will report them to the *American Oil Burner Association*.

No. 828

HEATING WITH STEAM BELOW ATMOSPHERIC PRESSURE

By C. A. THINN,[1] CHICAGO, ILL.

MEMBER

THE heating profession has long realized that there is a large amount of heat wasted in mild weather because buildings are overheated. The cause of this waste in steam heated systems is that most all of them are operated on and limited in their operation to pressures above atmospheric, even in mild weather. Steam pressures ranging from a few ounces to 2 or 3 lb afford but a small operating range of steam temperatures to meet changes in weather conditions. That is steam at relatively constant pressure is usually supplied to the radiators regardless of what the outside weather may be. The heat output of the radiators will, therefore, be entirely too high in mild weather, when only a fraction of the radiation installed would actually be required to balance the heat loss of the building and maintain the desired temperature. The building temperature will increase above that desired, resulting in excessive heat loss from the building structure. This is generally termed overheating and represents a direct waste of fuel.

In designing a heating system sufficient radiation must be provided to heat the building to a specified temperature during extreme weather. The base temperature for designing the radiation is selected accordingly, usually with an assumed operating pressure of 1 to 5 lb. This results in more radiation being installed than is needed for moderate weather and an excessive amount for mild weather. Consequently, with steam circulating on pressures at atmospheric or above, overheating occurs during both mild and moderate weather, if steps are not taken to control the loss. Government weather reports show that extreme weather prevails during about 5 per cent of the average heating season in Chicago, when the system must be operated at full rating. This leaves about 95 per cent of the heating season when the heat output of the system must be reduced if uniform room temperature is to be maintained.

Daily weather reports show that the temperature rises and falls quite rapidly over the greater portion of the country. Temperature changes of 40 F per day occur each winter in the Chicago area, while daily changes of 20 F are but slightly higher than normal. To this variation in heat demand must be added

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Presented at the Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, January, 1929.

the cooling effect of wind, if a true picture of the great fluctuations in heating requirements is wanted.

The graph, Fig. 1, shows the official maximum and minimum temperatures at Chicago for each day of the heating season of 1926-27. It clearly shows the temperature fluctuations for that season and also the extreme limits of temperature recorded in the previous 54 years. During that year, there was a total of 131 hours when the temperature was plus 10 deg or lower, with but 35 hours of this at zero or below. The heating season was but 0.4 deg warmer than the average since the year 1900.

This statement of the great variation in weather conditions from day to day is simply to point out that progress toward comfort and economy in heating should be made by making the heating system very flexible in heat output. The ideal in

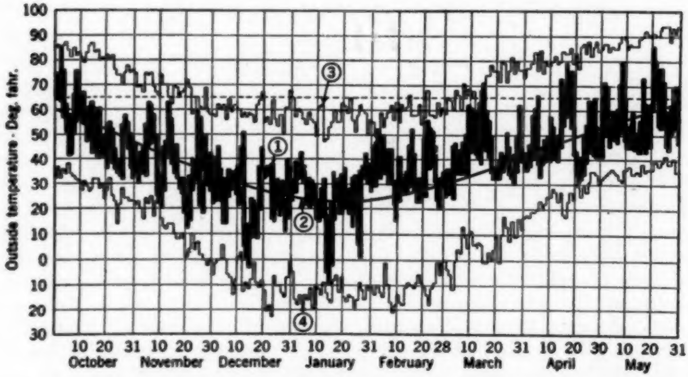

FIG. 1. AVERAGE TEMPERATURE FOR HEATING SEASON 1926-27=40.3 DEG

1 Maximum and Minimum Daily Temperatures for 1926-27
2 Normal Daily Temperature from 54 Year Record
3 Highest Temperature on Record
4 Lowest Temperature on Record

heating is to have a heat output of the system just equal to the heat loss from the building if uniform room temperature is to be maintained and heat waste through overheating, with the accompanying excessive open window loss is to prevented.

The available economies in heating fall roughly in two broad classes. The first economies are in heat generation which concern the boiler and the efficiency of combustion. That portion is not included in the scope of this paper. The second and by all means the largest field of possible saving is in heat utilization to which this discussion is confined.

Heating with steam at pressures below atmosphere is a simple way of obtaining flexibility in heat output while maintaining uniform room temperature, this steam being circulated in the piping system under a positive differential sufficient to cause flow through the radiation. By operating the heating system at pressures from atmospheric to as low as 2 or 3 lb absolute, the temperature of the steam circulated in the radiation may be varied from 212 F to about 130 F.

This will give a very large variation in the heat emission of each radiator. By varying the heat output of the system in this manner, it is quite easy to maintain uniform room temperatures.

The several parts of a system using steam at pressures less than atmosphere are very similar to the well-known vacuum return line heating system of which this is a further development. Fig. 2, disclosing a typical layout of a manually controlled system having a steam source with special reducing valves to supply steam at the desired temperature in the radiation, a properly designed system of steam piping, a regulating plate in the inlet of each radiator, a thermostatic trap on the outlet of each radiator, a system of return piping, a vacuum pump capable of producing a high vacuum, and an automatic electric switch. The switch is controlled by the pressure difference existing between the supply side and return piping.

Steam is supplied at the same vacuum (absolute pressure) at which it is to be used in the heating system by controlling the rate of heat generation at the boiler; this is the usual practice in smaller installations. On larger installations and on central station installations steam is furnished at a higher pressure than is required by the system and is reduced by means of pressure reducing valves to the desired vacuum or absolute pressure. These valves may either be manually or thermostatically controlled, one may be of sufficient capacity to care for mild weather operation and the other of sufficient capacity to care for cold weather, so that either or both may be used as heat requirements demand.

The piping is assembled so as to remain tight over a long period of years, just the same as is required for any other good job. The same quality of workmanship which is used to assemble the hot water supply piping is easily sufficient for all demands of a good heating system if proper provision is made for the slightly greater expansion.

As said previously, steam is used at pressures below atmosphere and its accompanying temperatures to give flexibility to the heat emission of the radiator. This pressure is varied at different times of the day and at different seasons of the year to maintain the room temperatures at the desired point.

Heating systems are seldom operated for 24 hours per day except in very severe weather, so the building is usually below the required temperature when the heating plant is started up each morning. The building may be brought up to temperature slowly or rapidly, as desired, by circulating steam at say a 10-in. vacuum or even on pressures above atmosphere just as the plant engineer may choose. When the building is up to the desired temperature the engineer can set the pressure reducing valve for the vacuum required to maintain the room temperature. The amount of steam to be supplied under this vacuum will, naturally, depend on the rate of heat loss from the building at the time.

By careful design and installation of a piping system, it is possible to get a distribution of steam at pressures which are fairly uniform throughout the piping. The use of a regulating plate with opening of proper size at each radiator inlet to assist in steam distribution at all three periods in the heating cycle will be observed. First, during the heating-up period, each unit should receive its portion of steam and no more; that is, a 25 sq ft radiator should heat at substantially the same rate as a 100 sq ft radiator. The regulating plates interpose a small resistance to the flow of steam and create a reservoir condition within the supply

piping, thus equalizing the pressure conditions at the entrance of each and every radiator.

During the heating-up period and while the radiators are filling with steam, the radiator traps are open to the return pipe and they remain open until steam reaches the trap when they close.

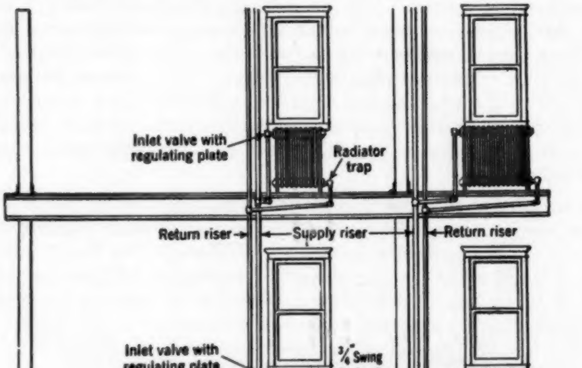

FIG. 2. TYPICAL LAYOUT OF A MANUALLY CONTROLLED SYSTEM FOR HEATING WITH STEAM BELOW ATMOSPHERIC PRESSURE

When the radiators are completely filled with steam and under normal operation, the regulating plates are intended to produce a semi-reservoir condition in the steam mains as compared with the condition when filling, by supplying an area for the flow of steam that is in proper relation to each size of radiator; if through some abnormal condition, the condensing rate of some radiator would be increased, the regulating plate will tend to prevent such a radiator from condens-

ing an excessive amount of steam, notwithstanding the demand for it. This is of value in reducing heat loss through excess window ventilation.

Radiators of a heating system completely filled with steam under varying degrees of vacuum meet the average heat loss required during a season, but early autumn and late spring days with very much higher outside temperatures,

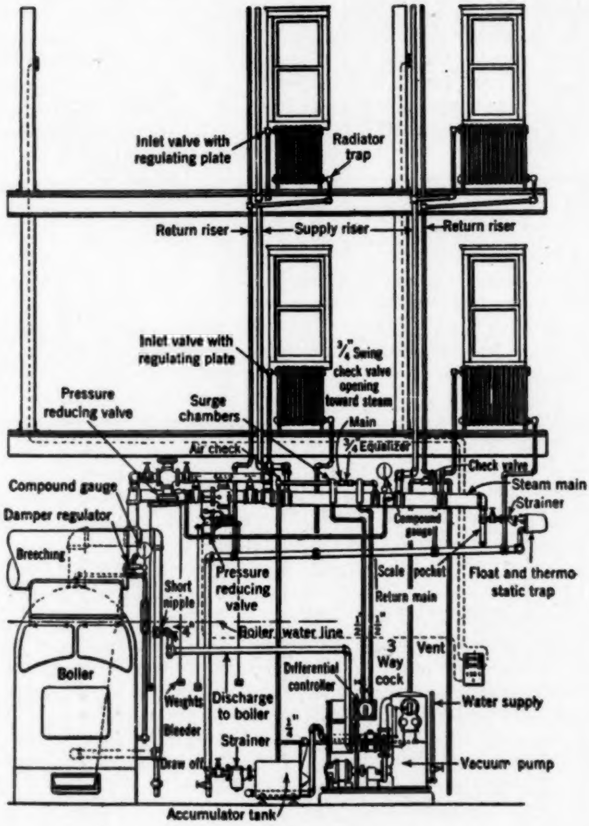

FIG. 3. TYPICAL LAYOUT OF AN AUTOMATICALLY CONTROLLED SYSTEM FOR HEATING WITH STEAM BELOW ATMOSPHERIC PRESSURE

require such a small heat output that even with a high vacuum, a radiator full of steam at a temperature corresponding to that pressure (vacuum) might under such conditions, overheat the building if the system was not capable of further heat reduction. If the amount of steam supplied to the system during such periods is sufficiently reduced and at the same time under a high vacuum, the

TABLE 1. DATA ON FUEL COSTS OF OFFICE BUILDING 1927-28

Month	Outside Temperature	Degree Days	Inside Temperature	Average Steam Main Pressure—In. Vacuum	Av. Hours Burner Operation	Fuel Oil Fired U. S. Gal 18-20 Baumé	Cost per Gal, Cents	Total Cost of Fuel, Dollars	Gal Oil per 1000 Sq Ft Radiation per Deg-Day	Condensation Meter Readings, Lb	Lb Steam per 1000 Sq Ft Radiation per Deg-Day
Oct.	58.0	236	71.1	18.78	6.25	1066	5¾	61.21	0.585	108,000	59.45
Nov.	43.2	655	70.0	18.62	11.2	2619	5¾	150.59	0.519	253,600	50.25
Dec.	26.9	1181	70.5	15.10	15.88	4599	5¾-5½	258.12	0.506	459,000	50.45
Jan.	25.2	1232	70.8	16.64	18.08	4881	5½	268.46	0.514	535,900	56.45
Feb.	30.3	1006	70.0	16.67	15.81	3640	5½-5¾	205.18	0.470	335,500	43.29
Mar.*	36.9	867	71.9	15.90	11.5	3575	5¾	205.56	0.535	335,500	50.25
Apr.	44.7	613	70.3	17.61	9.86	2143	5¾-5¼	120.85	0.454	210,800	44.64
May	58.2	232	71.7	16.20	3.47	679	5¼	35.65	0.380	63,000	35.24
Totals and Averages	40.4	6022	70.8	16.94	11.50	23,202	$0.0563	1,305.62	0.500	2,301,300	49.61

* All March plant data, except boiler pressure, includes a week of operation as a vacuum return line system.

TABLE 2. COST OF HEATING THE OFFICE BUILDING DURING HEATING SEASON, OCT. 1, 1927 TO MAY 31, 1928

Basis of Comparison	Actual Cost of Oil	Cost of Steam, Assume a Unit Cost of $1.00 per 1000 Lb
Total cost	$1305.62	$2301.30
Cost per sq ft of radiation	0.1695	0.2987
Cost per sq ft of rentable floor area	0.04298	0.07577
Cost per 1000 cu ft of total cubage	2.185	3.85
Fuel oil per sq ft radiation	3.012 U. S. Gal	
Cost of generating 1000 lb steam	$0.5675	
Pounds of steam per sq ft of radiation	298.8 lb.	

regulating plate will control the admission of steam to each radiator so that each remains partly filled, causing a further reduction in the heat output.

Next, it should be noted that a thermostatic radiator trap on the outlet of each radiator is of importance, for by its use a positive differential can be maintained between the pressure (vacuum) in the radiator and the pressure (vacuum) in the return pipe, thus securing satisfactory heating with a minimum amount of radiation. Steam circulation is maintained by the use of a jet exhauster type vacuum pump capable of producing a high vacuum. The pump motor is controlled by an automatic electric switch, which is actuated by the pressure difference in the supply and return piping. The function of this controller is to automatically start or stop the pump so that a small but substantially constant pressure difference is maintained. This differential will furnish a head sufficient to cause steam flow toward the returns.

The proper provision for the maintenance of this pressure difference is a most important factor in securing consistency and economy of heating; this point was brought out by C. A. Dunham in his paper given before this Society at the Annual Meeting in St. Louis in January, 1927. With properly designed piping, a differential of not exceeding 2 in. of mercury will keep the radiation drained of condensate on steam pressures below atmosphere and maintain steam circulation.

Even though it might be necessary to make changes in the radiation installed in a particular room to meet the desires of a permanent tenant who wishes room temperatures slightly above or below normal or because of partition changes on some floors, such changes will not bring the system out of balance by causing improper steam distribution elsewhere, because the steam mains and risers are used somewhat as a common reservoir from which the steam supply to each radiator is proportioned by means of a regulating plate at the inlet of each radiator. The use of properly sized plates alone will give a slight increase or decrease in the effective heating of any unit within certain limits.

Automatic control of the steam is frequently desirable in place of manual control which has just been described. Fig. 3 illustrates a typical layout of a heating system which gives automatic control of the steam supply.

Thermostats located in key rooms are used to control the operation of the larger of the two supply valves previously referred to. This valve, located directly in the steam main may be a motor operated valve. Its function is to admit additional steam to the heating system when the room thermostat calls for heat and to stop this flow when it is no longer required. The room thermostats are electrically connected to the motor of this valve through a control panel, arranged so that the steam in the system may be either automatically controlled or manually controlled when special conditions may demand it.

In buildings where varying wind conditions cause different demands for heat on the windward side, the use of several thermostats located on different sides of the building in separate key rooms has been the practice. The heating system may be under the control of any one of these thermostats if desired.

As buildings become larger or as the use of their several parts are more varied, it has been found that more economical means of satisfactory heating will be obtained by using a multiple system called zoning.

There are three main reasons why a large building should be divided into separate unit systems or zones. The first reason is to permit a higher steam and radiator temperature to be carried on those sides of a building which are tem-

porarily exposed to the then prevailing wind. By this means each zone can be maintained at a uniform temperature where otherwise it would be necessary to overheat, say the south or sun-warmed side.

The second reason is according to occupancy or the hours of use. Take the case of a building with stores or shops on the first and second floors, which

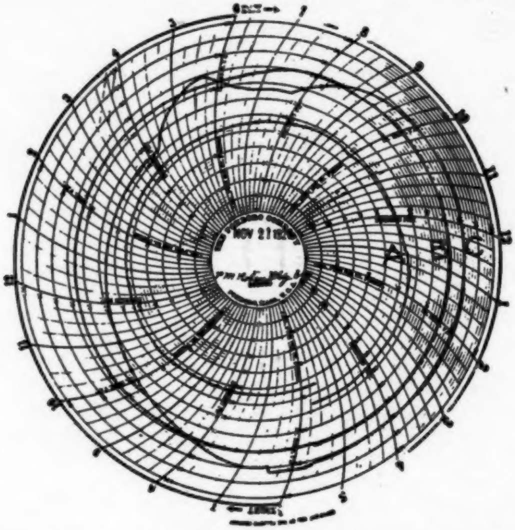

FIG. 4. PRESSURE AND TEMPERATURE RECORDER CHART FROM DUNHAM BUILDING, NOV. 21, 1928

A. Outside Temperature. B. Inside Temperature. C. Steam Main Pressure (Vacuum).

Heating load, 7704 sq ft direct radiation and hot water for wash rooms

Official temperatures (max 44 deg, min 32 deg)	Mean 38	F
Official prevailing wind S. W., average velocity	15.0	mph
Official maximum wind N. W., maximum velocity	27.0	mph
Oil burner operation (5:30 A.M. to 5:30 P.M.)	12	Hours
Fuel oil fired (18–20 deg Baumé)	109	U. S. Gal.
Gal oil per 1000 sq ft radiation per deg day	0.524	Gal
Flue gas temperature, average	225	F
Carbon dioxide in flue gas, average	9	Per Cent
Pump operation (4:20 A.M. to 6:30 P.M.)	14.16	Hours
Power consumed by pump, 17 kwh or	1.2	kwh per Hour
Condensation meter reading for 24 hours (6 P.M. to 6 P.M.)	11,500	lb
Pounds of steam per 1000 sq ft of radiation per deg day	55.29	lb

require heat for one period—say 7:30 A.M. to 7 P.M. with offices on the third to tenth floors which require heat for another period—say from 8 A.M. to 9 P.M. and with lodge or club rooms on the three upper floors requiring heat from noon to midnight. Here it is recommended that the heating system be divided into at least three zones, and if a much larger office portion were included, we would consider zoning that for wind and sun effects.

A third reason for zoning is to meet the different temperatures required. An excellent example is the common case of a factory building which has a shop section heated to 60 F, an office heated to 70 F, and a garage or warehouse heated to 50 F. Owing to the great and variable differences in ventilation requirements

of the three portions, especially with garages, it is difficult to secure uniform heating, simply by installing the proper amount of radiation. The use of one steam temperature for all zones at the same time is in this case incorrect because of the varying heating demand loads.

To secure satisfactory heating it is best practice, when conditions permit, to design the system by zones so different pressures and steam temperatures may be carried in each.

With such an installation the operating engineer should have instantaneous information before him so that he can know whether the temperature of each zone is at or below the desired amount at any time, and, from his central location, control the temperature of the steam which is circulated in each zone. This arrangement gives that properly controlled flexibility which is so essential to satisfactory heating with economy in steam consumption.

To present a true picture of the heat waste which is now taking place in systems where heat is not used in the most economical manner, permit us to quote from the *Proceedings* of the *National District Heating Association* for 1927, giving operating statistics for 1926 of the central heating companies reporting. Twenty-seven companies supplying steam to a total equivalent load of 30,641,000 sq ft direct radiation, reported an average steam consumption of 555 lb per sq ft per season. The seasonal temperature cannot be accurately determined from this record because three of these companies did not report on temperatures, however, the average of all the companies that reported temperature is 41.1 F. This is about 2 deg above the 54 year average at Chicago. In the year 1925 they reported an average steam consumption of 500 lb and in 1924, 537 lb per sq ft per season.

Nelson Thompson, chief mechanical and electrical engineer for the U. S. Treasury Department has presented to this Society in the July, 1928, issue of the JOURNAL, his method of estimating steam consumption for Federal Buildings located north of Richmond, Va. He "multiplies the square feet of direct steam heating surface by 500 to ascertain the number of pounds of steam condensed per annum in the system."

In Chicago the estimated figure of 500 to 600 lb of steam per square foot of direct radiation per season is a commonly accepted standard.

These figures on steam consumption, it is believed, are reliable averages. On this basis and from test experience, it is estimated that 450 lb per year represents good performance of a vacuum return line steam heating system in a Chicago office building and about 650 or even 700 lb per sq ft of radiation per year to be the performance of the more inefficient systems. No fixed figure can be set for the steam consumption of a system because of variation in building construction, heating system design and installation, hours of occupancy, nature of use, varying weather, etc. Therefore reliable comparisons can be made only by comparative test performed on the same building.

By use of steam at varying pressures below atmosphere, the Dunham Building in Chicago having 7704 sq ft of direct radiation was satisfactorily heated to an average temperature of 70.8 F during the past heating season (October 1, 1927 to May 31, 1928) with a steam consumption by meter of 298.8 lb of steam per square foot of radiation for the eight months. The average vacuum carried in the radiation was 16.9 in. which gives a corresponding steam temperature of 172.7 F. A base temperature of zero was used in calculating the radiation, so

the radiation installed is not in excess of actual needs. The building was fully occupied during the heating season and the tenants were permitted to make such use of window ventilation and control of radiator inlet valves as they desired.

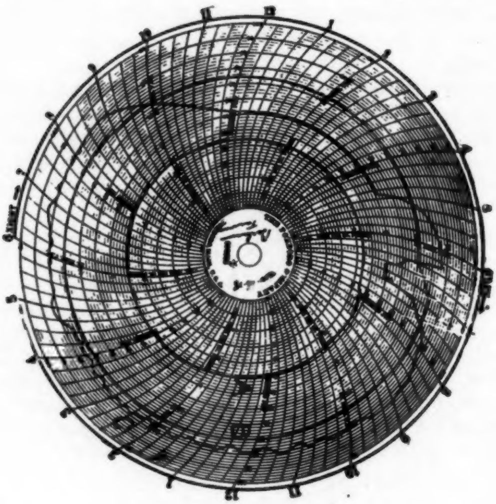

FIG. 5. PRESSURE AND TEMPERATURE RECORDER CHART FROM THE BARLUM TOWER, MAR. 16, 1928

A. Outside Temperature. B. Inside Temperature. C. Steam Main Pressure (Vacuum).

BARLUM TOWER DATA

Heating season, Sept. 12, 1927 to June 5, 1928, inclusive	
Average temperature, October to May, inclusive	39.9 deg
Total heating hours (for this building)	3258 hours
Total direct cast iron radiation installed	48,000 sq ft
Total radiation in use this period	40,800 sq ft
Total steam used during season	11,916,900 lb
Total cost of steam at $1.00 for 1000 lb	$11,916.90
Cost per sq ft of radiation in use	0.292
Cost per sq ft of rentable floor area	0 0396
Cost per 1000 cu ft of total cubage	2.684
Average cost per heating hour	3.657
Cost per sq ft of radiation in use per heating hour	0.00008965
Steam consumption for the heating period per	
Sq ft of radiation in use	292.08 lb
Sq ft of rentable floor area	39.62 lb
1000 cu ft of total cubage	2684 lb
Average steam consumption per heating hour total	3657.72 lb
Average steam consumption per sq ft of radiation in use per heating hour	0.08965 lb

The summary (Table 1) of the operating data by months gives the pertinent facts concerning this plant for the past heating season.

Table 2 gives a summary of the unit costs for this building computed in the several ways indicated. The total cubage and rentable floor area were computed in accordance with the code of the *National Association of Building Owners and Managers.* This cubage is the gross volume of the building measured as if the whole were a solid submerged in water.

Fig. 4 shows a pressure and temperature chart from a three pen recorder located in the boiler room. The thermometer elements of this recorder are of the distance type, the room temperature being that of the seventh floor which is used as a key room to indicate the building temperature. The outside thermometer bulb is located in a shelter on the north wall. This chart shows the cycle of heating for one day from 5:30 P.M. November 20 to 5:45 P.M. November 21, 1 28. The room temperature was maintained very uniform throughout the day on steam pressure of approximately 20 in. vacuum.

This heating system was operated as a vacuum return line system for one week in March, 1928, during which the outside temperature averaged 37.2 F, this was 3 below the season's average. The average consumption of steam for the year when operating with steam below atmosphere was over 33 per cent less than with a vacuum return line steam system during this test period. These steam rates were computed on the degree-day basis to correct for temperature differences.

Heating the Barlum Tower, a 40 story office building in Detroit, Mich., is an example of the use of steam below atmospheric pressures. This building was completed in September, 1927. Eighty-five per cent of the total 48,000 sq ft of direct radiation was turned on during the past season, giving 40,800 sq ft in use. Table with Fig. 5 gives a summary of the cost of heating this building. Steam was purchased from a central station heating company. The pressure and temperature chart from a recording instrument located on the 15th floor is shown in Fig. 5 to illustrate the pressure carried and temperature maintained with the outside temperature as indicated on the chart. This chart is self explanatory.

An industrial plant in Providence, R. I., having 6022 sq ft of equivalent radiation made a fuel saving of 33 1/3 per cent when a comparative test was made during two successive weeks when operating the plant with steam below atmosphere as compared with vacuum return line steam system operation.

A high school in Colorado made a saving of 38.8 per cent in tons of lignite coal required to heat the building with steam below atmospheric pressure during the past heating season as compared with a vacuum return line steam system in use the previous season. This school has a total of 4967 sq ft of direct radiation. In this and the previous cases, proper comparison has been made by reducing the data to fuel or steam used per degree-day.

A hotel in Tacoma, Washington, having 11,000 sq ft of direct radiation made a saving of $1428.00 in cost of central station steam used for heating during the first six months of 1928 by using steam at varying pressures below atmosphere as compared with the same period during 1927 when they used a vacuum return line steam system. The temperature being 0.9 deg higher during this period in 1928 than in 1927.

These figures reveal a few cases where savings have been made and the large buildings provide the same opportunity for savings as do the smaller buildings.

DISCUSSION

J. A. Donnelly (Written): This paper, as well as many others that have been presented before the Society, serves to illustrate the necessity of some authorized standard of comparison for checking the results produced or claimed.

The fundamental information necessary for a rational start on the problem has

to do with the actual heat loss of a building under some standard condition. This might be taken as the ascertained heat loss for zero weather, no sunshine and a 15-mile wind velocity. The equation for the steam required per heating degree would then be:

$$\frac{1,000,000}{1,000 \times 70} = 14.3 \text{ lb steam per hour per million Btu ascertained heat loss.}$$

This figure, if correctly ascertained, would serve as a standard for 100 per cent efficiency.

For other than this standard condition, we should, no doubt, need a great deal of research work on pro-rated heat losses, varying sunshine and wind velocity. In the case of buildings not continuously heated, allowance would have to be made for the intermittent heating-up of the building.

JOHN F. HALE: I remember in the early part of this century a building in this city heated by a vacuum system which was operated at about 12 in. of vacuum on the supply and about 15 in. of vacuum on the return, so what is before us today is not new except that it was not developed for practical use until many years later.

Again I recall in about 1904 a man by the name of Eugene F. Osborne developed two systems known as the simplex and the duplex systems, in which a vacuum was carried on the supply, and a greater vacuum on the return, making it possible for this refinement which is spoken of in the paper by Mr. Thinn.

I just wanted to call attention to this to show that in the days when Mr. Osborne was putting his system before the people, he was claimed to be 25 years ahead of his time, and if you will figure it up you will note that was in 1904, and today is 1929, so he was just 25 years ahead of his time.

R. V. FROST: There are one or two items in the paper that strike me as somewhat exaggerated. One is the statement of a saving of 33 1/3 per cent made during two successive weeks when operating the plant with steam below atmosphere, as compared with vacuum return steam heating operation in a high school and another where there was a saving of 38.8 per cent in fuel.

I would like to know if the tests were really on a comparable basis. We so often hear of these very high percentages of saving, that you question at once just under what conditions the tests were made.

It seems to be a common expression, that you can save 50 per cent of your fuel by simply a slight change. This seems to be a stock expression of different manufacturers. If we can make such a saving, between two different systems, then the old system must have been extremely inefficient, or else, perhaps, we can by multiplying the effect, heat our plants for nothing.

I would like to ask the speaker if he could explain to us a little more in detail just how these tests were operated, and if he is positively sure there was a 33 1/3 per cent saving in fuel.

We had an example in newspaper advertising, just a few weeks ago, in which it was claimed by a boiler manufacturer they had made on tests a saving of approximately 40 per cent in fuel. It was found afterwards that the tests had been improperly conducted, so there was not really the saving of 40 per cent in fuel. It would be almost impossible at any rate to choose two boilers at the present time in which one could be honestly said to be 40 per cent better than another. So in this case in comparing a system at pressure below atmosphere, with

a vacuum return line system, can there really be 38.8 per cent saving in fuel?

C. A. Thinn: The tests were in charge of an engineer, readings were taken every hour, and recording instruments were used at various points. The system when operating as a vacuum return line system was operated as is customary with such systems, keeping the pressure on the system near to atmosphere, probably a few ounces to a couple of pounds.

The control of the heat in the building was left to the occupants of the building, as is customary. Take this building here as an example, I would think that the engineer intends to keep enough steam at the radiator valves, leaving it to each occupant in his room to turn the valve on and off as he cares to.

However, that is just where the point is. The occupants are not interested in the operating cost of a building. Consequently, they fail to turn off their valves when the temperature reaches 70 F. The result is that the room temperature becomes 80 to 85 F before it is noticed, and the window is thrown open. So, although the room might show a reasonable temperature, the heat output of the radiation is increasing on account of the window waste. So that accounts for, in a great measure, why savings are possible when using steam below atmospheric, because with such a system the control of all the radiator valves in a building is taken out of the hands of every Tom, Dick and Harry in the building, and put into the hands of one man in the boiler room to give them just exactly the heat that they ought to have to maintain their rooms at 70 F.

The system, of course, is balanced up fairly well by means of the regulating plates, so that by reducing the temperature of the steam, the radiators will also reduce their heat output substantially on an even basis.

Go into buildings heated with sub-atmospheric steam and you will see very few windows open, probably 2 in. at the bottom to give a little ventilation. Go into buildings that are heated on pressure above atmosphere, leaving it to the occupants to turn the radiator valves, and you will find that perhaps 50 to 60 per cent of the windows are open at least one-third from the bottom. That condition represents a tremendous waste of heat.

The percentage of fuel saving was obtained by reducing all data to the degree-day method, which is, so far as I know, the only one which can be used in connection with these comparisons. I have not been able to run across anything that applies itself so readily.

Prof. F. E. Giesecke: In our institution, the Texas Agricultural and Mechanical College, there are two dormitories, each having 108 rooms in which we are conducting almost exactly the same investigations which Mr. Thinn has been conducting, with this difference: Mr. Thinn has studied the relative costs of operation of the ordinary steam heating system and of a special steam heating system in which a very high vacuum is maintained. We are studying the relative costs of operation of an ordinary gravity return steam system, and a hot water heating system. In our case, we feel that the cost of operating the circulating pump should be charged against the hot water system before the comparison with the steam system is made.

In Mr. Thinn's comparison, it may be necessary to find the cost of maintaining the high vacuum in the special heating system. I would like to know if this was done, and, if so, what the cost was in comparison with the cost of the heat supplied by the system.

MR. THINN: I could not answer the question directly because I do not have the figures, but the buildings that we have made comparison on were equipped with pumps. In other words, if we had the vacuum return line system, we would have a vacuum pump in the first place, and this pump would be operating from its regulator, that is, the regulator would be set at, say, 7 or 8 in. maximum, and 3 in. minimum. That gives the operating range. The pump would be running fairly well, as you probably know from your observations on vacuum return line systems.

When using sub-atmospheric pressure, the equipment which controls the pump is constructed in such a way that the pump will maintain, for instance, 2 in. of vacuum in excess of what would be on the steam line. For instance, if there was 20 in. of vacuum on the steam system, there should be 22 in. of vacuum on the return system to obtain circulation and get the steam into the radiators.

Under those circumstances, the pump would cut out as soon as 2 in. in excess of what you carried on your steam piping was reached and the pump will cut in and out in the same manner as with the vacuum return line system. It may run a little longer on the high degrees of vacuum than on low degrees. The system is not always working on 24 in. of vacuum. The average vacuum for the Dunham Building was about 16 in. of vacuum for the heating system. Today, for instance, as I left the office, there was 18 in. of vacuum on the system and the building was at about 72 F.

S. R. LEWIS: Was the boiler running under pressure or under vacuum?

MR. THINN: The boiler is under pressure and under vacuum.

Now, then, you realize that when a thermostat calls for heat, the reducing valve is going to open. That will, of course, take the pressure away from the boiler. However, the burner will begin to build up a supply because as soon as pressure is taken away from the boiler, the boiler control will open the oil burner to full capacity and it will take probably 5 or 10 min before the pressure is again built up on the boiler; then the regulator of the oil burner will reduce the oil flame. So we may say that it may be on pressure; it may also be on vacuum several times during the day, and the pressure is changed, of course, according to the way your heat is supplied into the system.

E. K. CAMPBELL: Is your oil burner controlled only by pressure of boiler?

MR. THINN: Yes, in this case it is; it is an industrial type burner. I know systems operating with this type of steam, where the burner is entirely automatic.

A. J. NESBITT: Have you ever had the opportunity of comparing the cost of a straight vacuum system equipped with pneumatic temperature regulation with the operating cost of the system advocated here?

MR. THINN: I never had an opportunity to do that.

GUY S. FABER: Having in mind the advantage of sub-atmospheric heating and listening to the speaker's paper, I want to ask him if the savings shown or the savings claimed for the system are not due primarily to temperature control or partly so. Perhaps part of the saving is due to sub-atmospheric condition, and part to temperature control or all to temperature control.

The paper cites a key room, or perhaps a zone part of the system. I have in mind a floor in a hotel of this type, in which you would perhaps pick out one room for key or zone room, and in that room you would locate a thermostat,

and that thermostat acting for the floor or wing of the building, whichever it might control, would be the key room.

Now, suppose that the occupants of that key room, not knowing that the room was a key room, or that a thermostat was located in it, should open their windows and necessarily create a greater heat demand. Would that necessarily overheat all the rest of the rooms where you depend entirely upon a key room to control a floor or a zone?

Mr. Thinn: In these buildings listed here, some are controlled manually and the results observed by means of a thermometer in the building while others are controlled from a thermostat, the thermostat controlling the reducing valve in the heating main. Of course, it all comes out in Btu. When you sit down and figure it out, it is heat output. Whether a man does it by using manual control or whether he does it by having an automatic equipment, is a matter of selection. You get better heating results by using thermostatic equipment.

You could probably shut off the steam in a building, say, eight or ten times a day, completely from a thermostat, and then turn it on again as soon as the temperature is below a given point. But interrupted heating is not really satisfactory heating, in my estimation. What we ought to strive for, is to give the tenants of a building continuous heat, by furnishing them the necessary number of Btu, to keep their rooms at the given temperature and do that all day long.

Now, all of us are acquainted with the steam table. The only trouble is that we have not looked at it long enough to find out that as soon as we begin to use steam below atmosphere, the steam temperature falls very rapidly. At the same time, the steam expands very rapidly in volume. This steam therefore lends itself ideally for furnishing the tenants the Btu that they ought to have, and at the same time, by this ability to expand into a great volume, fill every nook and corner of the system with steam.

Now, the trouble with this man, who was living 25 years ahead of his age, was that he could not take charge of this steam and control it, because equipment that would do this was not available. He did not have good traps and valves and regulating plates and pumps that go into a modern system of this kind.

The man who was 25 years ahead of his time had the right idea; why shouldn't he? The steam table has been before us for many, many years, but it had not been put to work using the lower temperatures, although there was a tremendous amount of overheating in the buildings. We are all trying to find ways of cutting down overheating.

F. Paul Anderson: I think we are under obligation to Mr. Thinn for this paper. It brings us some very interesting facts.

As a matter of fact, it requires very little more heat to make steam at 10 lb pressure than it does at atmospheric pressure so that it really requires very little more coal to make steam at the higher than at the lower pressure. Of course, we all know that the real amount of heat is in the latent heat of vaporization, which is about 971 Btu out of a total heat of 1151 Btu for atmospheric pressure and about 954 Btu out of a total heat of about 1162 Btu for 10 lb pressure. That radiation rates will be lower with low pressure steam than with high pressure is of course conceded.

Now, we must bear in mind, I think, in studying this subject of heating some of the psychological elements that enter into the whole question. I would dis-

like to see the time come when we would be fed by tablets. We would probably get all the calories and all the elements of food through the prescriptions of the great physicians, but I am sure it would be at times very pleasant to smell a big beefsteak, and to do some reckless eating. We would get then as we do now some of the satisfactions of living.

I think this paper is a worth while contribution from the standpoint of calling attention to the desirability of paying more attention to the balancing of our heating systems, and that radiation should be properly figured to secure balance. If radiation is not balanced, then diaphragms are prescribed. It seems to me there are certain types of buildings where we should not carry exact heating too far. Saving money all the time should not be the eternal objective of the engineer's life. Extravagant heating is desirable sometimes.

Now, we all like to get into a hotel room that is overheated sometimes, and recklessly keep up the windows. I would resent living in a place all the time where I would be bottled up and be afraid to let a little air in or I would be cold. For some kinds of heating, that might be entirely satisfactory. I think Mr. Nesbitt's question is very pertinent. It seems to me a very important consideration, now as always, is proper thermostatic control. Thermostatic control should be the keystone of heating economies.

This paper of Mr. Thinn's has a great value in pointing us to methods of exact heating. In certain types of buildings like hotels and residences for purely silly reasons to give humanity some variety we can profitably indulge in the frivolity of having too much heat. I think there is nothing quite so fine for dreaming as the old-fashioned fireplaces; certainly that is not an economical device for heating, so I believe in hotels and residences, and certain other types of buildings of conviviality it is well, sometimes, to consider in our heating engineering the matter of comfort as expressed in variation and excess.

Now, the advantage of the vacuum system just described, it seems to me, is not so much in the question of actually saving Btu as it is of getting a very complete circulation producing general heating service in a large building.

The paper is very interesting in that it shows us a scientific approach and a method of balancing our heating systems, but I think it is a question largely of automatic temperature control that we would save 30 per cent. I have no doubt that Mr. Thinn's figures are correct because this Edgewater Beach Hotel could be probably so revamped that at least 30 per cent of the fuel could be saved, but would lovers and other guests on pleasure bent think the rooms as cozy?

L. A. HARDING: This has developed into a very interesting discussion, and I have only two questions to ask Mr. Thinn. Would you not have obtained identically the same saving with a forced hot-water circulating system under proper control, and if so, what advantage has a sub-atmospheric system, with the possible exception of first cost of installation? Is not a steam system which operates under less than atmospheric pressure an attempt to equalize the performance of a forced hot-water circulating system?

MR. THINN: In regard to the question of advantage, I would say this: the advantage would be in the early and late part of the winter, the spring and the fall, when all we would want would be a little bit of heat.

With the forced hot-water circulating system, you would have over-heating during warm weather periods and changes in weather. Water is more sluggish

in circulation than steam. By producing high vacuum on the system, you will obtain steam quickly from the boiler. For instance, on a certain day the water in the boiler may begin to boil at 150 to 160 F because of the vacuum produced over the water. Consequently, steam is furnished very quickly and just enough to give the small amount of heat needed for that day.

W. B. Crawford: You are all heating experts and technical experts in one or more branches of the heating business but how many engineers have actually invented anything new in the heating art or have established a departure from standard practice? Nevertheless within the last two years there have been wonderful improvements made through the invention and application of various mechanical devices to the method of heating with sub-atmospheric steam. I am in absolute harmony with Mr. Thinn on the subject of sub-atmospheric heating.

Our organization has demonstrated this theory in actual practice on representative buildings, but we solve the problem from the standpoint of automatic control somewhat differently, but the results in savings are practically the same. I have an installation in mind, a 26 story building located on the Chicago Beach property, exposed on all sides. This building has approximately 40,000 sq ft of direct radiation and a water heating load of 3,000 gal per hour. The coal consumption has averaged 2 tons per 24 hours during the mild weather preceding the last cold spell and it has never exceeded 4 tons per day of 24 hours in the coldest weather. Temperatures on the lake front range as low as 14 F below zero.

We have been able to keep the domestic water at a temperature of 160 F with 10 in. of vacuum on the boiler, the direct radiation is supplied with steam at sub-atmospheric pressure of 15 in. of vacuum. We rarely go below 15 in. of vacuum on our system on account of the principle of the radiator trap which we employ, which makes the radiator inefficient in proportion to the vacuum.

At 15 in. of vacuum on the system the overall efficiency of the radiator will be approximately 70 per cent and the average temperature will be about 140 F. At atmospheric pressure the radiator efficiency will be about 99.3 per cent.

In a sub-atmospheric system of heating some means must be employed to keep a differential pressure on the return side of the system. This is accomplished by a vacuum controlling governor. The governor automatically keeps the pump working at a differential of 2 in. to 4 in. and when the boiler is working at atmospheric pressure or above the pump is automatically set back to standard conditions. This system worked out very well on a seventeen flat building with 100 per cent automatic control. The oil burner as well as the vacuum pump was automatic in operation.

The building has 3150 sq ft of direct radiation and a 1000-gal water load and the fuel oil has not exceeded a cost of $2.00 per day of 24 hours, except during the recent cold weather, when the cost per day for fuel oil was $3.00.

On the larger building mentioned first we estimate a saving of $8,000.00 this heating season against the estimated cost of heating this large building with an ordinary return line vacuum system. If the saving in fuel is an object and the comfort in the milder weather of the tenants is important, then the sub-atmospheric system of heating has ample justification for our consideration.

S. R. Lewis: It seems to me it is about time for us to form an organization of those who will agree that they will never put in another single-pipe system of steam heating to heat any place where people have to live.

DR. E. VERNON HILL: I do not think that Mr. Thinn defends his paper with sufficient aggressiveness. My knowledge of heating is not as broad as most of you gentlemen; I have never, however, been able to find out where there was any economy, theoretical economy, I mean, whether they use steam at 10 lb or vacuum or sub-vacuum. It seems to me that this is a new, and I think a very ingenious way of controlling the amount of steam to the various rooms, and I became interested in it at the Power Show in New York, and it struck me as something very worthwhile. I do not think that if you had a perfectly controlled system at a vacuum or at 10 lb, there would be any saving, but here is an ingenious and simple way of controlling the heat.

I cannot let Dean Anderson's remarks go unchallenged, however. He wants a superabundance of all things; but it seems to me that good engineering is marked from other things by its economy. If this method shows economy, that is the important thing, from an engineering standpoint.

MR. HALE: About 40 years ago there was a man in Philadelphia by the name of Napoleon W. Williames, who discovered that in trying to use exhaust steam for heating he was putting too much back pressure on his engine. He had the happy thought of connecting a vacuum pump to the end of the return so that it would remove as much of the atmospheric pressure as possible, and take off the back pressure on his engine. That was the origin of vacuum heating.

J. R. MCCOLL: Mr. Thinn has raised one point about which I would like to ask a question. He made the statement that it is hard to regulate with hot water. He also said 130 F was the minimum low temperature practical with steam. He could certainly get less than 130 F with hot water, and more evenly.

H. R. LINN: When Mr. Thinn brought up the heating with hot water by forced circulation, it took me back 20 years when I was in that business. I think Mr. Thinn's paper is a wonderful paper. Every now and then some member of this Society presents a paper like his, that starts us all to thinking. If we can only get men to thinking, by and by they will act. A few of us act when we do not think and then we are sorry afterwards.

However, hot water heating by forced circulation is not adaptable to all kinds of buildings. That is, it does not show the same efficiency in a hotel that it does in a factory or a school building or a hospital. In factory heating, it shows a much higher efficiency than it does in a hotel. If any one of you ever went into the hotel down in Bloomington, Ind.—I have forgotten the name of it—it was about 11 o'clock at night and the room was the way Dean Anderson wanted it, about 72 F, and the weather was zero outside; when you went to bed you did not sleep and a few minutes later you threw all the covers off but the sheet; along about 2 or 3 in the morning, that radiator had cooled off to the point where the next morning you did not talk any, because that room was heated by hot water with forced circulation.

That is one of the failures of forced hot water that you do not get with this sub-atmospheric heating; although I believe that in a factory building I would be willing to challenge the sub-vacuum heating on an efficiency basis. I am sure it would show as high an efficiency, if not higher.

I went into a hospital that was heated with forced hot water one day this fall, not a very cold day, but they needed a little heat and I found that the water was entering the radiator at about 90 F, just enough to take the chill off the room.

That could be done with thermostatic control on a radiator, or on manually controlled radiators for that matter, but I doubt if it is. My experience has not borne that out. However, I still think we have a lot to learn about this sub-atmospheric heating. Someone suggested that we ought to know more about the actual cost of operation. I think that is a very logical suggestion because in the end, if we pay a lot for saving a little, we will not have much in the bank to show for it. I would like to see some experiments carried on next year.

F. D. Mensing: If we will do a little thinking back on our own work, I think things relative to this paper will come to our minds. We have had experience with the *Paul system*. (It is no longer a patented system, so we will use the name.) We have had in connection with this system vacuums at the boiler. Our memories will show us that sub-atmospheric operations of steam system are not new and will work.

If we will check up our ordinary every day two-pipe vacuum systems, we find (unless there is an engine on the job) that at many times they work sub-atmospherically. We have a job where we have found as much as 19 in. of vacuum.

I would also suggest when we go into sub-atmospheric vacuum systems, we check up on our boilers and find what will be the result. Boilers are not designed for vacuums but for pressures. Vacuums are difficult to handle particularly in cylindrical vessels of large areas.

I have read this paper carefully, and I do not think any of us have enough data at hand to carefully check it. A saving of 35 per cent needs more to substantiate it to me than the figures given in this paper. I would not trust my eyes or those of anybody in my organization or anybody I know, to prove a 35 per cent saving on fuel. I would want automatic instruments that did not go asleep on the job.

I have had some bitter experiences along that line and I guess some of the other engineers have. Going away from our field for a moment, I will give you one experience. I was trying out a motor installation where we used the common ordinary everyday indicating type of instrument. It was very well checked and a 15 hp motor was decided on. About two years after we bought a recording instrument of a popular name, and I thought a good place to try out this instrument would be on this same job so we set it going. A red line, plus the oil pot in the bottom to act as a damper, showed we needed 3¾ hp.

Now, we did the best we could. I think our friend here has done the best he could, but now remember every Btu you put into a building, be it factory, apartment house or hotel, can only escape from that building in one way, that is, through the building structure, and unless you know the conditions of escape you cannot check the percentage of your saving.

In The Guide, we call for 15 per cent additional for north and west exposures, I believe. I may be wrong on that, but it is very close to that.

In the system as explained I do not see how the orifices can be changed with the change in the wind. Therefore, on two of the sides, at least, and maybe three, you are going to have overheating.

There is more to this paper than has been published. I think it is only right that those who have started this job should finish it.

R. G. Rosenbach: Mr. Hale made some remarks about the gentleman by the name of Osborne, who about 25 years ago conceived the idea of utilizing steam for heating purposes below atmospheric pressures. About 25 years ago, a great many industrial plants and other plants, operated electric power plants in which the engines were running condensing.

It was Mr. Osborne's thought that if the steam which was delivered to the condenser could be utilized in the heating system by being circulated through the heating plant that a considerable saving could be accomplished by making use of this steam or portions of this steam going to the condenser, instead of all being condensed in the condenser unit and the latent heat in this condensed steam wasted. It was usually the case that a condenser vacuum ranged from 25 to 28 in. Mr. Osborne developed the scheme of circulating steam at pressures of from 25 to 28 in. of vacuum through the heating system.

The principle of the idea was excellent and considerable savings were accomplished by utilizing the exhaust steam otherwise wasted into the condenser by circulating it through the heating system. It was soon found, however, that difficulties were experienced because the condenser vacuum gradually began to drop, resulting in the decrease of power output. This drop in the condenser vacuum was occasioned by the leaks developing in the piping system through the heating installation.

In spite of the efforts made to maintain the jobs tight under such extremely high vacuums, it was found almost impossible to keep them so and the maintenance costs of keeping the jobs tight to be suitable for such high vacuum conditions in the heating system were quite high and presented some complicated problems. The idea was therefore subsequently abandoned, as it was felt that it would not be practical from the standpoint of operation of the heating system, especially those of considerable size, to be able to keep them tight and prevent air from leaking into the system.

As already mentioned, savings were accomplished, but it was more important that the condenser efficiency should be maintained so as not to interfere with the power output of the prime movers. It was felt that it was not advisable from the standpoint of business policy to sell a client a job which was supposed to be circulated at high vacuum and later find that the man had to worry along with his installations with a vacuum considerably less than that offered him at the start.

You will note from the remarks that the difficulties encountered were solely due to the inability of maintaining a tight job. I would like to ask Mr. Thinn if he has had any difficulty in keeping his jobs tight after the second season, and I think that the other gentlemen would be very much interested to know what provisions he makes to keep the job tight under extremely high vacuum conditions, say, over 15 in., and what guarantee he thinks the heating contractor should give as to the tightness of the job and particularly as to the perpetuity of the tightness.

Mr. Thinn: In regard to the question of tightness of heating systems, when we first started this method of heating in the Dunham Building three years ago, we tested that system for tightness by obtaining 20 in. of vacuum on the system cold. It took three hours on that test to lose 10 in. of vacuum by leakage.

Then we ran the system for one season, and again we made a test on the

system with exactly the same kind of conditions as we had on the first test. We found that the system had become tighter. The fittings and the pipe thread, evidently must have rusted the joints tighter. We all know how difficult it is to tear down old piping.

What do the hot water circulation men have to say, who are depending upon tightness of system? What would take place in a building like this one with a forced hot water system, if it would not last more than two or three years? What would become of all the decorations, etc., if it should give out all of a sudden? Plumbing systems, for instance, are made tight and remain tight until the building is torn down or the pipes rust out.

In regard to the operating range, this system will operate on pressures as low as 25 in. of vacuum. Twenty-five inches of vacuum corresponds to 1½ lb absolute. There is not very much left on the gage. By referring to the paper you will find that it is possible to reduce the average radiator temperature below 130 F because steam under such high vacuum is not filling the radiators completely. You might have your radiator on very mild days cool at the bottom and warm at the top, say 130 F at the top. Now, if you figure out the averages, you will have an average radiator temperature of 80 to 90 F.

W. H. Carrier: I should hesitate to enter into this discussion, since my business practice has been largely with air heating, were it not for the fact that a personal experience with the heating system in my home leads me to disagree with the previous discussion of Mr. Lewis relative to the general undesirability of one pipe steam systems.

A year ago I would have decided with him very pronouncedly because I had a one pipe heating system in my house that was the worst thing that was ever installed. However, since then I have made some changes in the type of radiation giving a better distribution and, what was more important I put in vacuum type air valves of a wellknown make and automatic control.

I have never had a finer heating system. I think it is largely the way one applies equipment whether the results are to be praised or condemned. I have changed my ideas completely as to the possibility of one-pipe systems doing effective heating, especially in the home. I do not know that I would recommend it for office buildings; that is a different proposition, but for the home the one-pipe system can be made to give very efficient results if properly installed, and equipped with the right accessories. Unfortunately, my system is not vacuum tight, although I went to considerable expense in having the valves, repacked, but I made a mistake in not putting in packless valves.

Speaking of leakage in piping systems, I think 99 per cent of that is in leakage at the valves. You can tighten the valve packing, but because of alternating pressure and vacuum they will not stay tight. The money I spent in repacking my valves was thrown away. I could have spent a little more and installed packless valves which would have remained perfectly tight. I would say that packless valves are a necessity with a vacuum system.

However, I believe that the ultimate system of the future will combine the advantages of both hot water and steam. It will be designed like a hot water system with few accessories placed not on the radiators but in the basement. It will employ a forced circulation of steam and air throughout the system. It will afford uniform distribution of heat to all radiators at all temperatures, just as will hot water. The quantity of heat will be easily controlled by orifices or

modulating valves. Vacuum type valves will be unnecessary because the pressure will always be at atmospheric or slightly above. At full load only steam and condensate will be circulated, while at other times a mixture of vapor and air at atmospheric pressure will be circulated. This will permit the desired variations in the temperature of the heating medium just as with hot water. At the same time it will give a more positive circulation and distribution of heat and will require less heating surface and give a more ready response than the hot water system.

I believe that such a system is feasible and will eventually be adopted. It offers the solution for many of the present difficulties in steam and hot water heating. The other solution is to use air directly as the heating medium.

A. J. DICKEY: There are several observations that I have made on this paper. One question was asked in connection with the value, I judge it was, of temperature control on radiators. It seems to me that we might reasonably expect that with this system and with all of the radiators equipped with individual temperature control, the system functioning at the low temperatures, there is bound to be a saving from losses which might otherwise occur in the opening of windows over what loss you would have if that system was operating at atmospheric pressure because of the high temperatures in the radiators. Would that be reasonable?

MR. NESBITT: If you are addressing the question to me, I believe it is principally a matter of regulation—if there is a saving in generating steam at lower temperatures it has not been established. It, therefore, seems evident to me that the regulation of the temperature is the principal advantage of this system.

MR. DICKEY: I misunderstood your question. Dean Anderson spoke of liking to have the hotel room with lots of heat. We have always looked forward to Dean Anderson's visit to Toronto, and when he runs out of any of those hot hotels down here, we can show him a lot of them up there. He will not have any trouble finding a hot berth.

This other feature, too, appeals to me, in connection with this system: That there is not a great deal of difference in the type of equipment, in connection with this sub-atmospheric system as described, than we have in the old-fashioned vacuum return systems. We have traps and pumps and piping arrangements much the same, and the application of the differential system idea, with the equipment that is required, does not mean a radical change in the application so that we have very little trouble individually or otherwise to prove out the thoughts that are expressed in this paper.

MR. THINN: Dean Anderson mentioned that the comfort of people should be considered. I have always learned that people are in comfort when their rooms are kept at a reasonable temperature. Doctors tell us that we should not have over 70 F in our homes; that if we overheat our rooms, we are apt to go outside and catch cold. If this overheating can be stopped, I should think that this would tend to improve people's comfort and their health. Furthermore, anyone who has been working alongside of a radiator filled with steam at 2 lb pressure steam knows what radiant heat is. Then contrast the difference between this and the same radiator operating at about 130 to 140 F temperature from which you will hardly notice any radiant heat, and you will see what is meant by comfort in a building heated by this type of heating.

No. 829

RATIO OF OPENING OF FAN PERFORMANCE IN TERMS OF DIRECT PRESSURE-QUANTITY RELATIONS[1]

By G. E. McElroy,[2] Pittsburgh, Pa.

NON-MEMBER

DUE to the large number of variables involved, there are many methods of presenting fan performance characteristics. Tabular data are in general use and serve fairly well, but they do not compare with graphic presentation of the required data for conciseness, for completeness, or for giving a real conception of relative changes in fan performance.

The simplest graphic method of presenting such data is to plot pressures, powers, and efficiencies against quantities on rectangular coordinates, for definite speeds and at standard air conditions. For single installations a graph of this type ordinarily satisfies all requirements and should be supplied by the manufacturer without request, and certainly on request.

For the general purpose of fan selection, a graph for every speed, size and type of fan would be impracticable and for this purpose many types of ratio charts are in use. These involve ratios of performance data rather than specific data, and apply to a whole line of fan sizes of a single type. Their use involves considerable mathematical ability, some degree of experience in using the particular type involved, and often some basic starting data not marked on the charts but given in supplementary tables. Some types are easier to follow and require fewer computations than others.

The writer has been particularly interested in mine fans for which the available performance data are in the form of ratio charts in which performance ratios are plotted against ratio of opening. The latter is the ratio that the area of an orifice on the discharge end of a short test duct bears to the area of the duct which is made the same as that of the fan discharge. The writer has also been particularly interested in the determination and calculation of direct pressure-quantity relations for mines and mine airways. The expression of ratio of opening in terms of direct pressure-quantity relations was therefore desired, and this involved the consideration of actual pressure-losses at orifices.

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[1] Printed by permission of the Director, U. S. Bureau of Mines. (Not subject to copyright.)
[2] Mining engineer, U. S. Bureau of Mines.

Presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Chicago, January, 1929.

Orifice Pressure Losses

The measurement of air flow in ducts by means of small pressure differences at inserted orifice plates is standard practice in industry and one that has been the subject of numerous investigations. These investigations show that the main difficulty in the practical application of the method is the fact that variable coefficients are involved, and a rigid selection from experimental data is required, based on the exact conditions of the orifice as to type, location and ratio of area to duct area. Practically all of the available experimental data on air flow relate to either thin-plate, sharp-edged orifices or special forms, and are therefore not applicable to the type of orifice used in determining fan performance, since the latter is generally a thick-plate, square-edged orifice symmetrically placed on the end of a pipe discharging into the atmosphere.

No experimental data for this type of orifice and condition of placement appeared to be available, so in the fall of 1926, in connection with some fan test experiments[3] conducted at Butte, Montana, in cooperation with the Anaconda Copper Mining Co., the writer determined pressure losses for four square-edged circular orifice plates, made of 7/32-in. basswood, installed on the discharge end of an 18-in. diameter smooth-iron pipe. Pressure losses for various positions of a cone orifice, used as a regulator for varying resistance conditions, were also determined in this same series of experiments.

New Constant for Expressing Orifice Pressure Losses

In analyzing these pressure-loss results, it was found that the static pressure at the orifice on the end of a pipe, for both the plate and cone types, bore an almost constant ratio to the difference in velocity pressure as between the orifice area and the pipe area. The constancy of the ratio was much greater for the cone orifice than for the orifice plates due, it is thought, to lack of sufficient precision in measuring orifice diameters. Confirmatory data for the latter type of orifice were therefore desired. These have been supplied, fortunately, by more recent experiments[4] of Weeks and Berray and of Callen and Mitchell on thick-plate, square-edged, square orifices inserted in square ducts. By assuming that the difference in pressure loss is represented by the difference in shock loss for the two conditions of installation, that is, using coefficients of contraction based on shock loss, it is possible to compute ratios of static pressure (SP) to differential velocity pressure (DVP) for the orifices of these experiments as used on the end of a duct. Similar ratios have also been computed from the data given in the handbook[5] *Fan Engineering* for thin-plate sharp-edged orifices on the end of a duct (p. 65) and from the coefficients given for the same type of orifice inserted in a duct (p. 71).

Values of the ratio $\frac{SP}{DVP}$ were computed from the shock-loss coefficient of contraction, termed "coefficient of discharge" in *Fan Engineering,* by means of a formula developed as follows:

Let N = ratio of duct area to orifice area

c = shock-loss coefficient of contraction

[3] McElroy, G. E., and Richardson, A. S., Experiments on Mine Fan Performance, Technical Paper No. 447, by U. S. Bureau of Mines.

[4] Weeks, Walter S., The Air-Current Regulator, and discussion; presented at the February, 1928, meeting of the *Inst. of Min. & Met. Engrs.*

[5] Fan Engineering, 2nd ed., 1925.

VP = velocity pressure in duct
SP = static pressure in duct at orifice discharging to the atmosphere
DVP = difference in velocity pressure as between orifice area and duct area.

The total pressure loss at an orifice discharging to atmosphere is $\left(\frac{N}{c}\right)^2 VP$, and the static pressure at the orifice is one duct velocity pressure less than the total pressure loss, or $SP = \left[\left(\frac{N}{c}\right)^2 - 1\right] VP$. The velocity pressure for the duct erea is VP and for the orifice area $N^2 \times VP$, therefore $DVP = (N^2 - 1)\ VP$,

$$\text{and } \frac{SP}{DVP} = \frac{\left[\left(\frac{N}{c}\right)^2 - 1\right] VP}{(N^2 - 1)\ VP} = \frac{\left(\frac{N}{c}\right)^2 - 1}{N^2 - 1}$$

Determined and computed values of the ratio $\frac{SP}{DVP}$ for orifices at the end of a duct and of the shock-loss coefficient of contraction (coefficient of discharge) are given in Table 1, using the above notation. The uniformity of the ratio is striking, particularly in the range of 0.2 ($N = 5.0$) to 0.7 ($N = 1.43$) orifice area ratio for which the probable accuracy of experimental results is greatest, and suggests the possibility that the lack of agreement for very low and very high orifice ratios may be due to experimental errors.

That accuracy is difficult to obtain experimentally at extreme orifice ratios is common knowledge and it is possible that calculated coefficients, based on SP/DVP ratios, may be more accurate for these extreme ratios than experimentally determined values. In any event, the practical range of orifice ratios is limited to 0.1 to 0.7, and it would seem that the SP/DVP ratio at least offers a method of securing faired-up values of the shock-loss coefficient of contraction, or coefficient of discharge, within this range.

A tentative conclusion that may be drawn from the general agreement of values of ratios derived from tests for two different conditions of installation of two different shapes of orifices of the same general type is, that, within practical limits of experiment and application, the pressure loss at orifices can be regarded as entirely due to shock-loss and independent of the shape of the orifice when the latter is symmetrical with the duct. This conclusion requires, of course, to be substantiated by more conclusive data.

Type Constant for Thick-Plate, Square-Edged Orifices

For the present purpose of determining what "ratio of opening" means in terms of pressure-quantity relations, the thick-plate, square-edged orifice data only need be considered; and, since fan performances at extreme ratios of opening have no practical importance, minor discrepancies for these ratios may be neglected. Inspection of the experimental results shown in Table 1 indicates that a good average value of the SP/DVP ratio for this type of orifice is 2.45. The experimental coefficients of contraction listed in Table 1 are plotted against orifice ratio in Fig. 1 in conjunction with a curve for the coefficient derived by calculation from $SP = 2.45\ DVP$. Inspection shows that the curve fits the

experimental data with a fair degree of accuracy and certainly with sufficient accuracy to permit its use in calculating approximate average data for the resistance of orifices used in fan tests.

Direct-Pressure-Quantity Relations

The relation of pressure to quantity, or resistance to flow, of a ventilating

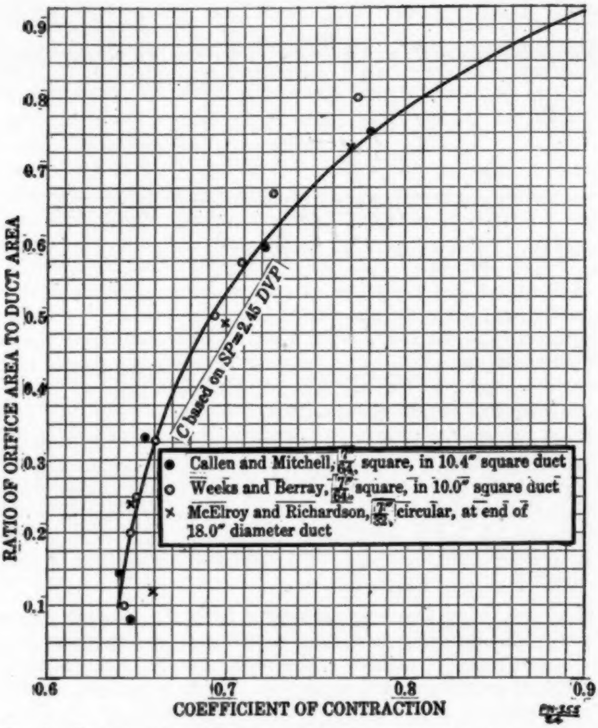

FIG. 1. COMPARISON OF EXPERIMENTAL DATA FOR THE SHOCK-LOSS COEFFICIENT OF CONTRACTION OF SQUARE-EDGED, THICK-PLATE ORIFICES WITH COEFFICIENTS BASED ON THE FORMULA $SP = X$ DVP; WHERE SP = STATIC PRESSURE AT ORIFICE ON END OF DUCT, X = CONSTANT FOR TYPE OF ORIFICE AND DVP = DIFFERENCE IN VELOCITY PRESSURE AS BETWEEN ORIFICE AREA AND PIPE AREA

system may be expressed, with a practical degree of accuracy, by the formula $H = RQ^2$, where H is pressure, Q is quantity of flow and R is a factor of specific resistance similar to the ohm of electrical resistance. For convenience of calculation and application, the author has found it to be desirable in mine ventilation practice to use pressure in inches of water and rate of flow in hundred thousands of cubic feet per minute. With these units, R is termed

TABLE 1. RATIO OF STATIC PRESSURE TO DIFFERENTIAL VELOCITY PRESSURE FOR ORIFICES AT THE END OF A DUCT AND SHOCK-LOSS COEFFICIENT OF CONTRACTION FOR VARIOUS TYPES OF ORIFICES

Authority		McElroy and Richardson				Weeks and Berray		Callen and Mitchell		Fan Engineering			
Position		End				Inserted		Inserted		End		Inserted	
Shape		Circular				Square				Circular			
Type		20° Cone				Thick-plate, square-edged				Thin-plate, sharp-edged			
N	$\frac{1}{N}$	$\frac{SP}{DVP}$	c	$\frac{SP}{DVP}$	c	$\frac{SP}{DVP}$	c	$\frac{SP}{DVP}$	c	$\frac{SP}{DVP}$	c	$\frac{SP}{DVP}$	c
12.00	0.083							2.40	0.647				
10.00	0.100	0.48	1.435			2.42	0.644			2.76	0.604	2.81	0.610
8.27	0.121			2.32	0.659								
6.75	0.148							2.46	0.641				
5.00	0.200	0.53	1.350			2.45	0.647			2.74	0.612	2.71	0.616
4.15	0.241			2.47	0.648								
4.00	0.250					2.46	0.650						
3.33	0.300	0.52	1.331							2.72	0.625	2.68	0.629
3.00	0.333					2.45	0.661	2.50	0.655				
2.50	0.400	0.51	1.303							2.69	0.643	2.72	0.642
2.05	0.487			2.39	0.699								
2.00	0.500	0.51	1.257			2.44	0.694			2.67	0.666	2.69	0.664
1.75	0.572					2.46	0.709						
1.69	0.592							2.41	0.722				
1.67	0.600	0.52	1.200							2.67	0.695	2.70	0.692
1.50	0.667					2.61	0.727						
1.43	0.700									2.72	0.730	2.74	0.728
1.37	0.732			2.47	0.770								
1.33	0.750							2.45	0.781				
1.25	0.800	0.62	1.076			2.86	0.774			2.82	0.777		
1.11	0.900									3.20	0.840		

"resistance factor" and represents numerically the pressure in inches of water required to maintain a flow of 100,000 cfm.

With the test duct on a fan of the same area as the fan discharge, the difference between total pressure (TP) and static pressure for both orifice and fan is one duct velocity pressure, and the static pressure at the orifice is the same as that at the fan, except that the latter is increased a small amount by the resistance of the test pipe. The latter is a negligible factor at small orifice ratio, but of some importance at high orifice ratios. It is necessary, therefore, to determine values of R in $H = RQ^2$ for both orifice and pipe.

Ratio of Opening in Terms of R

Since $DVP = (N^2 - 1)\ VP$, $SP = 2.45\ (N^2 - 1)\ VP$ and $TP = [2.45\ (N^2 - 1) + 1]\ VP$, pressure losses for any orifice ratio can thus be computed in terms of duct velocity pressures. To determine values of R for orifice ratios it is necessary to express VP in terms of R.

Let VP = velocity pressure in inches of water
V = velocity in feet per minute
q = quantity in cubic feet per minute
d = air density in pounds per cubic foot
A = area in square feet
Q = quantity in hundred thousands of cubic feet per minute.

The standard formula for velocity from velocity pressure is

$$V = 1097 \sqrt{\frac{VP}{d}},$$

and by transposing,

$$VP = V^2 \times d \times 0.000000831.$$

At standard air density of 0.075 lb per cu ft this reduces to

$$VP = 0.0000000623\ V^2$$

But

$$V^2 = \frac{q^2}{A^2}$$

and by substitution

$$VP = \frac{0.0000000623\ q^2}{A^2} = \frac{623}{A^2} Q^2.$$

For a pressure equivalent to X velocity pressures, then,

$$H = \frac{623\ X}{A^2} Q_2, \text{ and } R = \frac{623\ X}{A^2}$$

Since X is constant at $\frac{N^2}{C^2}$ for constant orifice ratio or ratio of opening, it is apparent that R will vary inversely with the square of the area of the duct and therefore of the fan discharge area.

Values of R for a length of one diameter of duct can be determined in the same form from the standard air flow formula:

$$H = \frac{KS}{5.2\ A^3} q^2 = \frac{KS}{5.2\ A^3} 10^{10}\ Q^2,$$

where K = friction factor; S = rubbing surface in square feet = length times perimeter; and H, A, q and Q have the same designations as before.

For smooth straight duct, the value of K at moderate to high velocities will average about 0.0000000012 or $\frac{12}{10^{10}}$. If D = diameter, then

$$R = \frac{10^{10} \text{ x } 12 \text{ x } D \text{ x } \pi D}{10^{10} \text{ x } 5.2\ A^3} = \frac{12 \text{ x } 4\ A}{5.2 \text{ x } A^3} = \frac{9.23}{A^2}$$

Values of R for both orifice pressures and duct pressures can thus be expressed in the form $R = \frac{C}{A^2}$ where C is a constant that can be calculated for either orifices alone or, by addition, for orifices and duct of test installations. Assuming ten diameters of duct for the average length of test pipe, values of the constant C in $R = \frac{C}{A^2}$ have been computed on a $SP = 2.45\ DVP$ basis, and are tabulated in Table 2 as representative of a typical test installation.

TABLE 2. VALUE OF RESISTANCE CONSTANT C IN $R = \frac{C}{A^2}$ FOR RATIOS OF OPENING BASED ON $\frac{SP}{DVP}$ RATIO OF 2.45

Ratio of Opening	Pressure Loss at Orifice in Velocity Pressures		Value of C			
			For Orifice Only		For Orifice Plus 10 Dia. of Pipe	
	Static	Total	Static	Total	Static	Total
0.10	242.5	243.5	151,078	151,701	151,170	151,793
0.20	58.8	59.8	36,632	37,255	36,724	37,347
0.30	24.78	25.78	15,438	16,061	15,530	16,153
0.40	12.86	13.86	8,011	8,634	8,103	8,726
0.50	7.35	8.35	4,579	5,202	4,671	5,294
0.60	4.36	5.36	2,716	3,339	2,808	3,431
0.70	2.553	3.553	1,591	2,214	1,683	2,306
0.80	1.377	2.377	858	1,481	950	1,573
0.90	0.574	1.574	358	981	450	1,073
1.00	0.000	1.000	000	623	92	715

Ratio of Opening in Terms of Pressure Ratios

In practice, although fan performance may be plotted against ratio of opening as a base, test results are more often based on pressure ratios whose relation to ratio of opening has been fixed by previous test work. Ratios of velocity pressure to static pressure may be used, although ratios of velocity pressure to total pressure are more common. Values of both, on the same basis as above, have been computed as the reciprocals of the pressures expressed in velocity pressures, and are tabulated in Table 3.

Ratios of velocity pressure to total pressure corresponding to the same ratios of opening, as taken from small-scale graphs of fan performance, are shown in

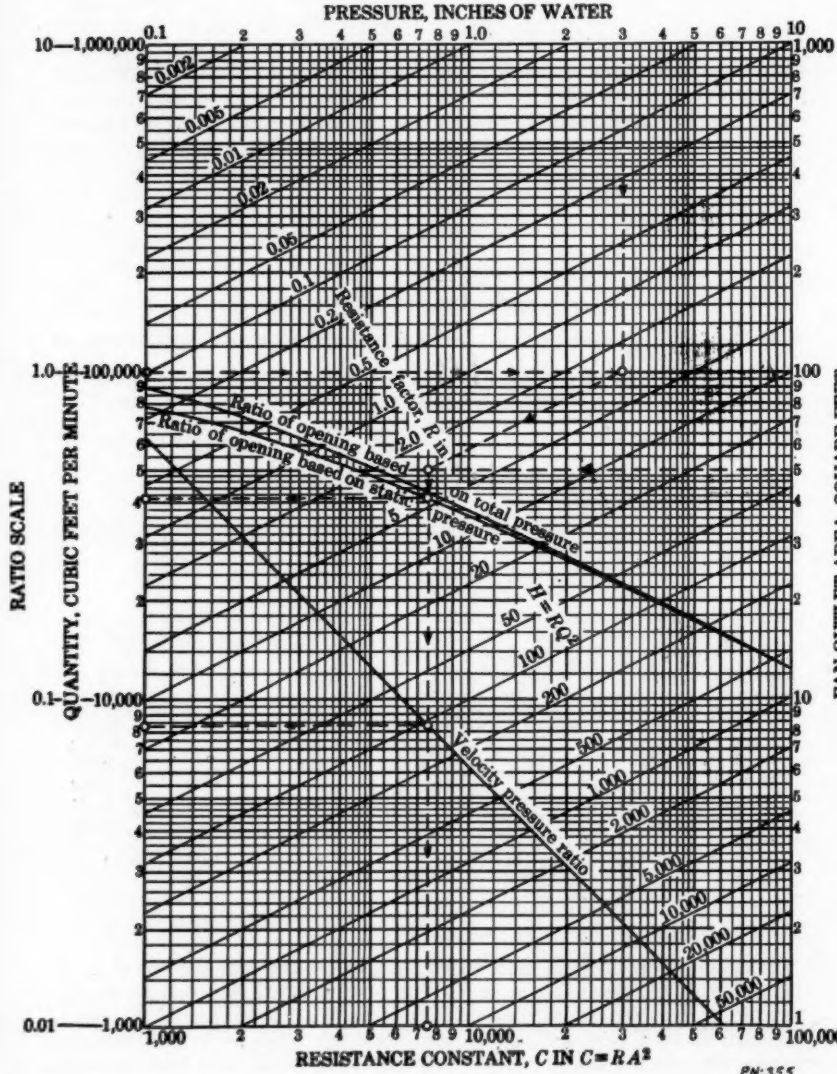

FIG. 2. CHART FOR THE RAPID GRAPHICAL DETERMINATION OF RATIO OF OPENING AND VELOCITY-PRESSURE RATIOS OF FAN PERFORMANCE FROM DIRECT PRESSURE-VOLUME RELATIONS OF VENTILATING SYSTEMS

the last three columns of Table 3. Apparently the test duct effect is eliminated in these values by adjusting the relation to ratio of opening to come out smoothly at 1.0 for a ratio opening of 1.0. Those in Column A are for a line of fans frequently encountered in mine ventilation and show good agreement with the calculated values. The figures in Columns B and C were selected at random from small fan tests and the higher values indicate that they were determined with orifices having rounded edges. Similar data for exhaust fan performance,

TABLE 3. VELOCITY-PRESSURE RATIOS FOR RATIOS OF OPENING BASED ON $\frac{SP}{DVP}$ RATIO OF 2.45

Ratio of Opening	Ratio of Velocity Pressure to Static Pressure		Ratio of Velocity Pressure to Total Pressure				
			Computed		Manufacturer's Data		
	Orifice only	Orifice plus 10 Diam. of Pipe	Orifice only	Orifice plus 10 Diam. of Pipe	A	B	C
0.10	0.004	0.004	0.004	0.004	0.008	0.015	0.010
0.20	0.017	0.017	0.017	0.017	0.020	0.033	0.035
0.30	0.040	0.040	0.039	0.039	0.046	0.058	0.078
0.40	0.078	0.077	0.072	0.071	0.079	0.098	0.140
0.50	0.136	0.133	0.120	0.118	0.132	0.150	0.220
0.60	0.229	0.222	0.187	0.182	0.200	0.227	0.340
0.70	0.392	0.370	0.281	0.270	0.287	0.355	0.475
0.80	0.726	0.656	0.421	0.396	0.425	0.535	0.660
0.90	1.742	1.385	0.635	0.580	0.647	0.755	0.800
1.00	....	6.780	1.000	0.872	1.000	1.000	1.000

actually based on static pressure at fan inlet, as would be expected, also show a general lack of agreement.

In general, then, while a good correlation of resistance factor to ratio of opening may be obtained for general application, it will undoubtedly fall down in special cases. In any case, however, in which the ratio of velocity pressure to total pressure is given, the value of the constant C on a total-pressure basis will be the reciprocal of this ratio times 623; and where ratio of velocity pressure to static pressure is given, the value of C on a static pressure basis will be equal in value to the reciprocal of this ratio times 623. On a total-pressure basis the normal value of C is 623 greater than it is on a static-pressure basis.

Chart for Determining Ratio of Opening

In order to determine the ratio of opening for a particular pressure-quantity relation—that is, for a particular value of R—the area of fan discharge must be known. In Fig. 2 the writer presents a chart on logarithmic coordinates for the rapid graphic determination of ratio of opening (based on thick-plate square-edged orifices) velocity-pressure ratios, resistance constant C, area of fan discharge, resistance factor R, total pressure, static pressure and quantity of flow in terms of each other. The scale for the latter is given, as a matter of convenience, in cubic feet per minute, although the actual values of R and C are based on quantities in hundred thousands of cubic feet per minute.

The general method of using the chart (Fig. 2) may be explained by the solution of the problem shown by the broken lines. In this problem the quantity of flow is 100,000 cfm; static pressure at fan discharge (blowing), corrected

to standard air density of 0.075 lb per cu ft, is 3.0 in.; and the area of the fan discharge is 50 sq ft. Referring to the chart: The intersection of the horizontal 100,000 quantity line with the vertical 3.0 pressure line establishes a point on the resistance-factor line (actual value 3.0) whose slope is shown by the diagonal 1 to 2 ruling. The intersection of this sloping line with the horizontal area line 50 establishes the value of the resistance constant C, read on the scale vertically below this point, as 7500. The intersection of the vertical resistance-constant line 7500 with the curve for ratio of opening based on static pressure establishes the value for ratio of opening, read on the outer left scale, at 0.41 (41 per cent), and its intersection with the velocity-pressure ratio curve establishes the value of the latter, on the same scale, at 0.083.

If total pressure were used in this problem instead of static pressure, $TP = 3.25$, $R = 3.25$ and $C = 8123$. The intersection of the vertical through the latter with the ratio of opening curve based on total pressure gives the same ratio of opening, 0.41, but a lower velocity-pressure ratio, as it should.

Charts for Fan Selection

A chart similar to Fig. 2 might be used to present fan performance and facilitate fan selection—that is, a ratio chart using the resistance constant C as a basic scale against which to plot performance ratios. The value of this constant in terms of direct pressure-quantity relations involves the area of discharge, but for general purposes of fan selection it is the value of C that is desired, since for the same value of C all sizes in the line are considered to have the same efficiency and operating characteristics. In selecting a fan, the value of R is known, and if the value of C for best efficiency for the fan type is known, the solution for area of discharge is simple. Area of discharge is usually related to the number specifying the fan size in such a way that the value of C may be modified to give a solution direct in terms of fan size. Actually, in making an economical fan selection, it is desirable to obtain, with a minimum of effort, complete data on several sizes of several types in order to arrive at a minimum total cost for both installation and operation. With no less than seven main variables involved, a much clearer and simpler presentation of fan data is desirable than is at present available for the general run of fans.

Ratio of opening and other ratios used as basic scales for presenting fan data have no fundamental basis, and it is well known that rectangular coordinates give no real data as to comparative rates of change between quantitites of unequal magnitude. What is required for presenting fan performance, in the author's opinion, is the more general use of logarithmic-coordinate plotting paper, so that equations involving powers and roots will plot as straight lines, and the selection of a base for plotting that involves resistance to flow in direct terms. In the latter connection it is suggested that values of C in $H = \frac{C}{A^2} Q^2$ might be used and termed "resistance constant" of fan performance.

What is really required for the simplification of fan selection is the use of charts that one may enter with definite quantity and pressure values or their equivalent, a resistance value, and come out with definite size, speed, power and efficiency data. There is no doubt that this can be done, but there is some doubt as to whether it can be done with a practical degree of simplicity. In order to secure maximum advantage from any form of chart, it is imperative

that a standard form be used by all manufacturers. Such a result could, of course, only be brought about through cooperation of the parties interested and could logically be handled through the *National Association of Fan Manufacturers* and the American Society of Heating and Ventilating Engineers.

Summary and Conclusions

In attempting to correlate ratio of opening used as a basis for expressing relative fan performance, the author has analyzed the results of pressure-loss experiments on orifices, with particular reference to the thick-plate, square-edged type involved in fan test work, and has found that, within practical limits of application, a single constant can be used to define a type of orifice. The constant referred to is the ratio of static pressure at the orifice, when discharging into the atmosphere, to the difference in velocity pressures between the duct area and the orifice area. The use of this constant is proposed as a method of fairing-up experimentally determined coefficients of discharge for orifices.

The constant derived from experiments on thick-plate, square-edged orifices, 2.45, has been used in calculating values of ratios of opening, and velocity-pressure ratios, in terms of direct pressure-quantity relations. This involves a new conception—termed "resistance constant"—for the expression of pressure-quantity-area relations and one that seems particularly adapted for use as a basic scale in presenting data on relative fan performance. A chart, based on this new conception, is presented for the rapid graphical determination of ratio of orifice, and velocity-pressure ratios from direct pressure-quantity relations.

No. 830

HEAT AND MOISTURE LOSSES FROM THE HUMAN BODY AND THEIR RELATION TO AIR CONDITIONING PROBLEMS

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MEMBERS

An understanding of the relation of man to his atmospheric environment is necessary for progress in the art of heating and ventilation. This principle has long been recognized by leaders in this branch of the engineering profession, and was one of the main factors leading up to the establishment of the Research Laboratory of the American Society of Heating and Ventilating Engineers.

Since its establishment, the Laboratory, in cooperation with the U. S. Bureau of Mines, has had under investigation as one of its most important lines of research, the relation between temperature, humidity and air motion and feeling of warmth, comfort, health and the physiological reactions experienced by human beings. This study has resulted in the development of information and data of great value to the public in general, as well as to the heating and ventilating engineer, the physiologist and the physician.

As a result of completion of another phase of this investigation, there is presented in this report, information concerning the rate of heat production in, and dissipation from the body, both for still and moving air. Heat loss from the body is differentiated into loss by radiation and convection or sensible heat loss, and loss by evaporation of moisture or latent heat loss. A number of curves for solution of practical problems in air conditioning and examples in their use are also presented.

To satisfy the processes of life, man takes into his system food consisting largely of oxidizable material, air containing oxygen as its essential constituent, and water. Through the process of metabolism, carbon, hydrogen and other elements in minute quantities, contained in the food, unite with oxygen from the inspired air developing energy for internal and external work and heat for maintaining body temperature.

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Presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Chicago, 1929.

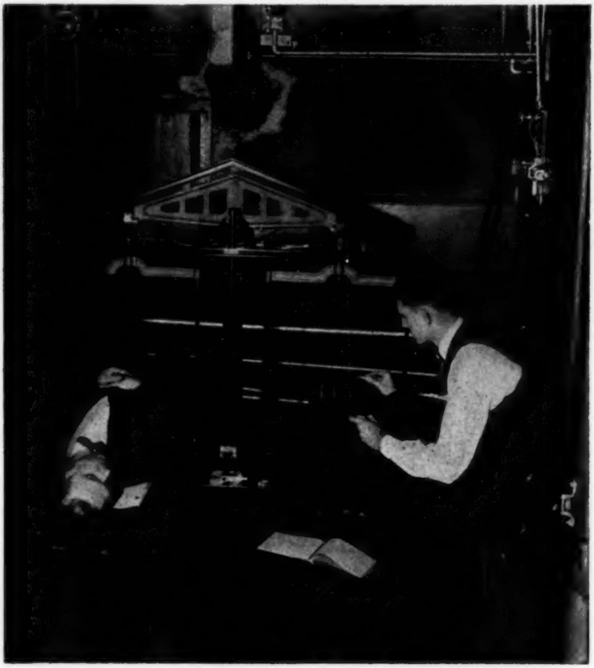

FIG. 1. BALANCE ON WHICH SUBJECTS ARE WEIGHED

The chemical reactions for the oxidation of typical classes of food in the body are exothermic or proceed with an evolution of heat. These reactions are stated by MacLeod[1] as:

1. Carbohydrate:

$$C_6H_{12}O_6 + 6O_2 = 6CO_2 + 6H_2O$$
(Dextrose)

2. Fat:

$$C_3H_5(C_{18}H_{33}O_2)_3 + 80O_2 = 57CO_2 + 52H_2O$$
(Olein)

3. Protein:

$$C_{72}H_{112}N_{18}O_{22}S + 77O_2 = 63CO_2 + 38H_2O + 9CO(NH_2)_2 + SO_3$$
(Albumin)

The calorific values used in determining the heat produced in the body for each of the three classes of foods are those universally accepted by the physiologist. Computations were actually based upon the volume of oxygen consumed rather than upon the weight of food. The values used for the three classes of foods are:

1. Carbohydrate:

567 Btu per cu ft of O_2 consumed or 6789 Btu per lb of dextrose.

[1] See Bibliography, p. 266.

2. Fat:

   526 Btu per cu ft of O_2 consumed or 17,114 Btu per lb of olein.

3. Protein:

   504 Btu per cu ft of O_2 consumed or 8646 Btu per lb of albumin.

That there is a change in total heat production with high and low environmental temperatures has been shown by a number of investigators. Possibly the most complete evidence of this fact is given by McConnell, Yaglou and Fulton[2] who found that basal metabolism was a minimum at effective temperatures of 75 to 83 F. There was a definite increase at temperatures above and below this range.

Krogh[3] shows an increase in metabolism for warm blooded animals at both low and high temperatures. Lusk[4] also shows data on warm blooded animals which give increased metabolism at high and low temperatures. Voit[5] and Rubner[6] give data showing an increase in heat production in man at low temperatures. Wolpert[7] also demonstrates this fact and goes further to show that wind velocity causes an additional increase. Hill[8] found that metabolism was much higher sitting out of doors especially in cold windy weather. He also

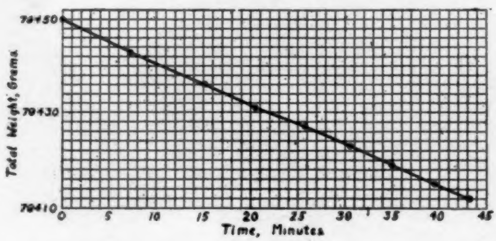

FIG. 2. EXAMPLE OF WEIGHT LOSS OVER TIME INTERVAL

states that Lefevre found the heat production at 5 C (41 F) was twice as great as at 20 C (68 F) with 1 meter per second (199 fpm) air velocity. Benedict, Benedict, and Du Bois[9] exposed men and women to a blast of hot dry air driven through an oilcloth bag in which the subjects were enclosed. Their heat production was increased by 5 or 10 per cent over that in room air.

For temperatures below those used in ventilation, the energy produced in the body is only sufficient to take care of the processes of life, work performed and to meet normal heat losses. As the temperature of the surrounding atmosphere rises above this level and the difference in temperature between the body and air becomes less, control of heat dissipation in order to maintain temperature equilibrium becomes the main function of life, or the adjustment of the internal to the external.

Under these conditions equilibrium is largely maintained through the availability of perspiration for evaporation. A large quantity of water vapor is added to the air through evaporation of perspiration from the skin and clothing, and moisture from the respiratory tract. A better understanding of these functions is of value to the heating and ventilating engineer for two reasons. *First,* a more complete knowledge of the relation of the atmospheric environment to man, his

[2, 3, 4, 5, 6, 7, 8, 9] See Bibliography, p. 266, 267.

comfort and physiological reactions will make it possible for the engineer to produce better atmospheric conditions for human comfort. *Second,* a better understanding of the effect of these processes of life on the surrounding atmosphere as regards addition of heat and moisture is necessary in designing heating and

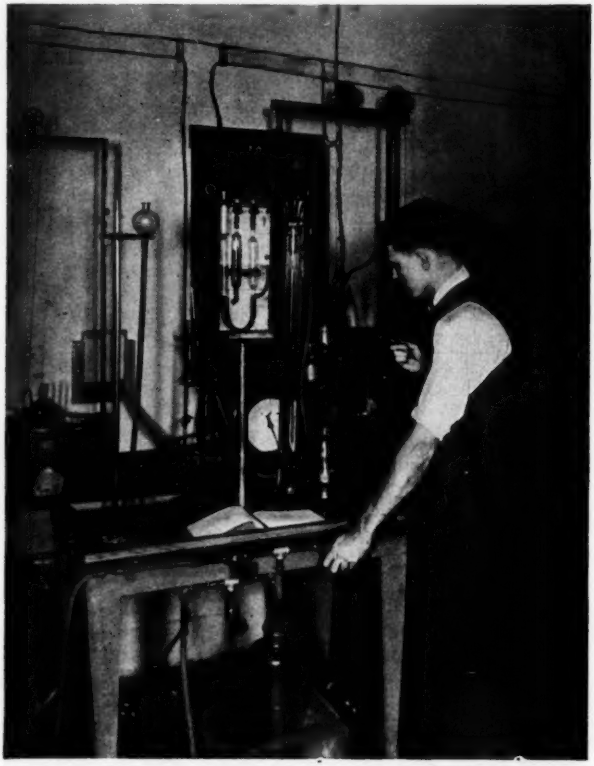

FIG. 3. METABOLISM APPARATUS FOR DETERMINING RATE AND ANALYSIS OF EXHALED BREATH

ventilating systems which will maintain desired atmospheric conditions in space occupied by large audiences.

The object and need of this particular phase of the more general problem of determining the relation between temperature, humidity and air motion and the feeling of warmth, comfort and health of human beings were outlined by the Technical Advisory Committee on this subject, under the chairmanship of W. H. Carrier.[10]

The purpose of the Laboratory investigation was to determine the rate of heat production in the body, the rate of heat dissipation to the air and the differentia-

[10] See Bibliography, p. 267.

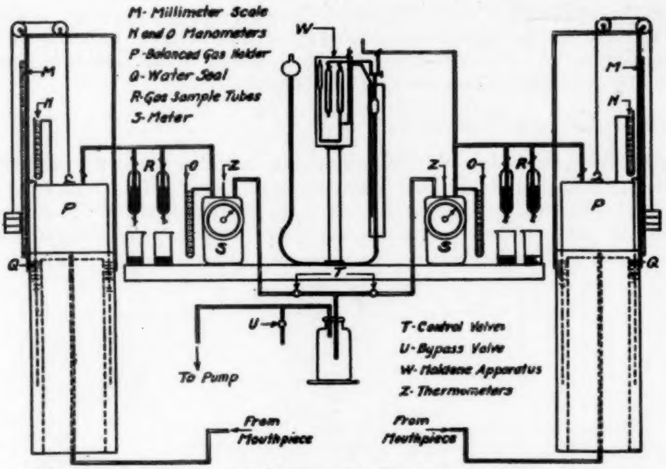

FIG. 4. APPARATUS FOR MEASURING, SAMPLING AND ANALYZING EXHALED BREATH

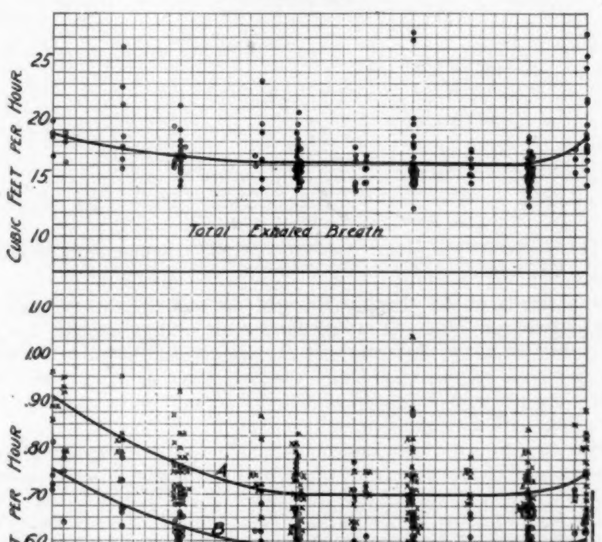

FIG. 5. RELATION BETWEEN EXHALATION, OXYGEN CONSUMED, CO_2 PRODUCED AND EFFECTIVE TEMPERATURE

tion of this loss between that taking place by radiation and convection, or sensible heat loss and loss by evaporation of moisture, or loss of latent heat. It was also the purpose of the investigation to determine the rate of addition of moisture to the air by evaporation from the skin and lungs.

Test Procedure and Observations

In this study 267 tests were made on subjects in the psychometric chambers of the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in the Pittsburgh Experiment Station of the U. S. Bureau of Mines. In either of these two rooms, which are completely described in another report,[11] any desired atmospheric condition may be maintained.

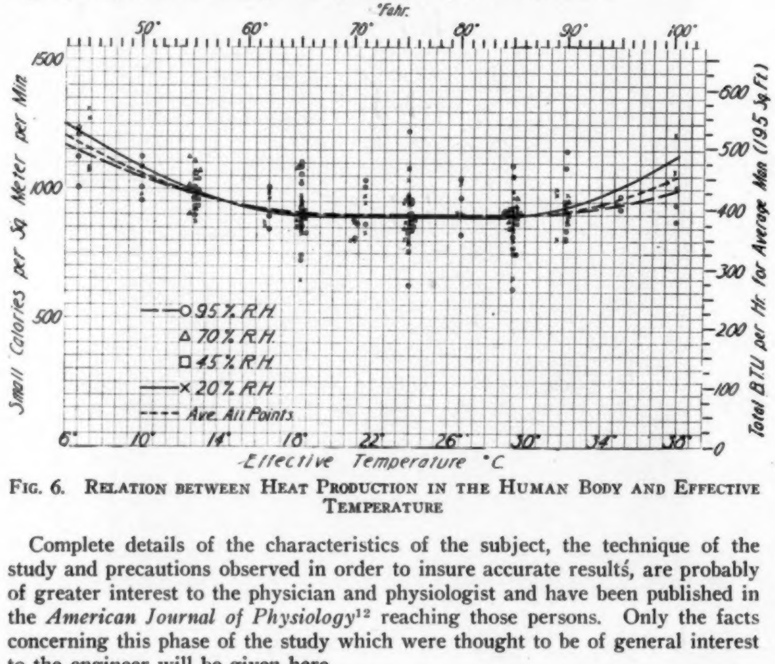

FIG. 6. RELATION BETWEEN HEAT PRODUCTION IN THE HUMAN BODY AND EFFECTIVE TEMPERATURE

Complete details of the characteristics of the subject, the technique of the study and precautions observed in order to insure accurate results, are probably of greater interest to the physician and physiologist and have been published in the *American Journal of Physiology*[12] reaching those persons. Only the facts concerning this phase of the study which were thought to be of general interest to the engineer will be given here.

The subjects were university students of good health and average physique between the ages of 19 and 24 years. Since heat production depends largely upon the activity and diet of the subject and since it was desired to have the data apply to persons normally at ease as in an audience hall, their ordinary diet and routine of life were not interfered with outside of the 4½ hours of the preliminary and test periods. During the preliminary and the 4-hour test periods, they sat at a table in a test chamber kept at the desired atmospheric condition. They were normally clothed, and their activity consisted of making a few test observations, and reading, studying or talking as was their pleasure. Unusual or strenu-

[11], [12] See Bibliography, p. 267.

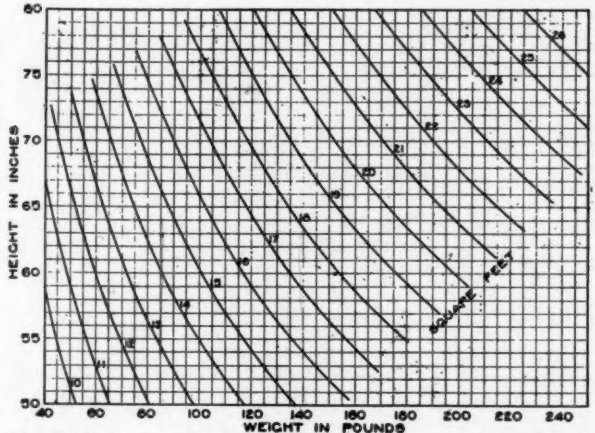

FIG. 7. CHART FOR DETERMINING SURFACE AREA OF INDIVIDUALS, HEIGHT AND WEIGHT GIVEN

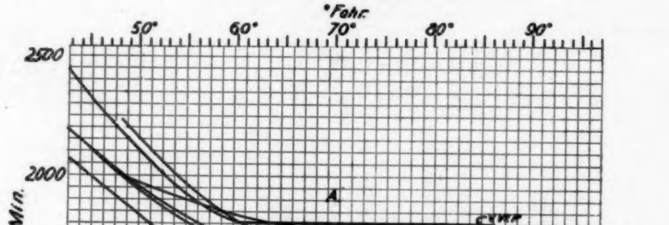

FIG. 8. RELATION BETWEEN INDIVIDUAL HEAT LOSS CURVES AND EFFECTIVE TEMPERATURE

TABLE 1. SAMPLE OF TEST DATA AND COMPUTATIONS FOR DETERMINING RATE OF HEAT PRODUCTION IN THE BODY, RATE OF HEAT LOSS BY RADIATION AND CONVECTION AND RATE OF HEAT LOSS BY EVAPORATION (*Note: Numbers in parentheses refer to items.*)

1. Test No. 11 — Date 2-1-27
2. Subject O. D. R. — Height 171.4 cm
3. Air condition 21.1 C (70.0 F) E. T.; 26.2 C (79 F) D. B.; 13.2 C (55.8 F) W. B.; 20 per cent R. H.
4. Rise in body temperature during the test period .... −0.17 C (0.3 F)
5. Time of temperature rise .... 150 min
6. Rate of temperature rise .... −0.0011 C/min
7. Actual weight .... 55.8 kg (123.0 lb)
8. Total weight loss during test period .... 209 grams
9. Weight loss other than C exhaled and H_2O evaporated .... 0 grams
10. Net weight loss, C exhaled and H_2O evaporated .... 209 grams
11. Time of weight loss .... 205.57 min
12. Rate of weight loss .... 1.017 grams/min
13. Volume of exhalation (as read on meter) .... 857.8 L (30.28 cu ft)
14. Volume of exhalation (corrected for meter calibration) .... 871.5 L (30.76 cu ft)
15. Time of exhalation .... 94.77 min
16. Rate of exhalation .... 9.19 L/min
17. Pressure of measured exhalation above barometric .... −8 mm H_2O
18. Pressure of measured exhalation above barometric .... −0.6 mm Hg
19. Barometric pressure .... 746.1 mm Hg
20. Temperature of measured exhalation .... 25.9 C (78.6 F)
21. Vapor tension of water at (20) .... 24.8 mm Hg
22. Absolute pressure of exhalation (19) + (18) − (21) = 746.1 − 0.6 − 24.8 = 720.7 mm Hg
23. Volume dry exhalation in liters per minute under standard conditions =

$$(16) \times \frac{273}{273 + (20)} \times \frac{(22)}{760} =$$

$$9.19 \times \frac{273}{273 + 25.9} \times \frac{720.7}{760}$$

.... = 7.96 L/min

Analysis of exhalation, per cent volume

24. CO_2 .... 3.13
25. O_2 .... 17.31
26. N_2 .... 79.56
27. Volume O_2 consumed per minute = (23) x

$$\left(\frac{20.93 \times (26)}{79.03} - (25)\right) \div 100$$

20.93 = per cent O_2 in normal air

79.03 = per cent N_2 in normal air 7.96 x

$$\left(\frac{20.93 \times 79.56}{79.03} - 17.31\right) \div 100 = 0.301 \text{ L/min}$$

28. Volume CO_2 produced per min = (23) x [(24) − 0.04] ÷ 100
0.04 = per cent CO_2 in normal air 7.96 x (3.13 − 0.04) ÷ 100 = ... 0.246 L/min
29. Respiratory quotient $= \frac{(28)}{(27)} = \frac{0.246}{0.301} = 0.82$
30. Small calories heat produced per Liter O_2 consumed 4777*
31. Heat produced by subject = (30) x (27) = 4777 x 0.301 = ... 1438 cal/min
32. Storage = (6) x (7) x 0.83 x 1000 = − 0.0011 x 55.8 x 0.83 x 1000 = ... −51 cal/min
0.83 = specific heat of the human body
33. Total heat lost by subject = (31) − (32) = 1438 + 51 = ... 1489 cal/min
34. Weight of O_2 consumed = (27) x 1.428 = 0.301 x 1.428 = ... 0.430 gram/min
1.428 gram = weight of 1 liter of oxygen
35. Weight of CO_2 exhaled = (28) x 1.965 = 0.246 x 1.965 = ... 0.483 gram/min
1.965 gram = 1 liter CO_2
36. Weight loss by evaporation = (12) + (34) − (35) = 1.017 + 0.430 − 0.483 = 0.964
37. 575 = Latent heat of water at 37.3 C (99 F) body temperature.
38. Heat loss by evaporation (36) x (37) = 0.964 x 575 = ... 554 cal/min
39. Heat loss by radiation and convection (33) − (38) = 1489 − 554 = ... 935 cal/min
40. Per cent heat loss by evaporation =

$$\frac{(38)}{(33)} \times 100 = \frac{554}{1489} \times 100 = \ldots\ldots 37.2$$

41. Per cent heat loss by radiation and convection $= \frac{(39)}{(33)} \times 100 = \frac{935}{1489} \times 100 = 62.8$

Pounds = Kilograms x 2.205
Cubic feet = liters x 0.03532
British thermal units (Btu) = Small calories x 0.003968.
Btu per hour for average man (19.5 sq ft) = small calories per sq meter per min x 0.431
Grains per hour for average man (19.5 sq ft) = small calories per sq meter per min (loss by evaporation) x 2.915

* Taken as 1% lower than values in Zuntz's table for non-protein respiratory quotient. 10–20 per cent calories from protein was assumed.[13]
[13] See Bibliography, p. 267.

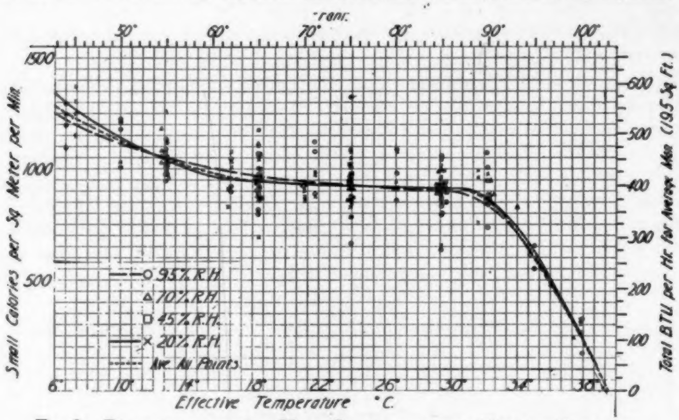

FIG 9. RELATION BETWEEN HEAT LOSS FROM THE HUMAN BODY AND EFFECTIVE TEMPERATURE

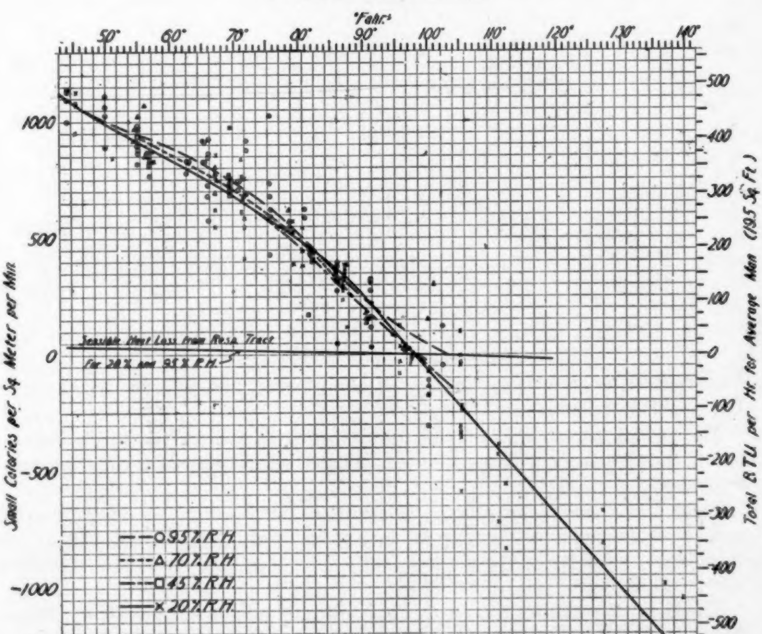

FIG. 10. RELATION BETWEEN HEAT LOSS BY RADIATION AND CONVECTION FROM THE HUMAN BODY AND DRY-BULB TEMPERATURE

ous activity was not permitted, but normal movement of body and limbs was not interfered with.

The data collected during the 4-hour test period constituted a heat balance test not unlike a heat balance test on a boiler. Weighing on the bullion balance, Fig. 1, gave the rate of weight loss of the subject to within plus or minus 0.2 gram. Fig. 2 shows the accuracy which could be obtained by frequent weighing. The weight loss is due to evaporation of moisture from the body and to the carbon dioxide exhaled less the oxygen consumed. Measurement and analysis of the exhaled breath with the metabolism apparatus, Figs. 3 and 4 gave the rate at which carbon dioxide was produced and oxygen consumed. The total rates of carbon dioxide production, oxygen consumed and exhalation in cubic feet per

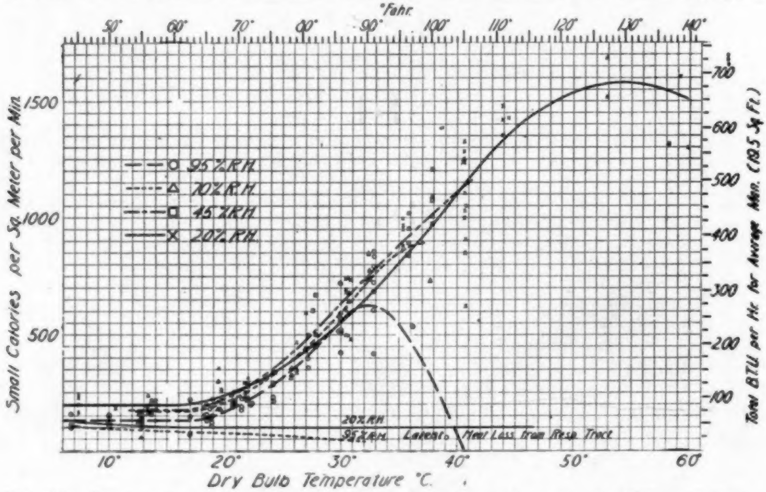

FIG. 11. RELATION BETWEEN HEAT LOSS BY EVAPORATION FROM THE HUMAN BODY AND DRY-BULB TEMPERATURE

hour reduced to standard conditions of temperature, and pressure are given in Fig. 5 for all tests made on the subjects.

The rate of heat production was determined from the analysis of the exhaled breath in accordance with an accepted method[14] of calculation giving the metabolic rate.

Table 1 is a sample set of data, calculations and results from one of the 267 tests. For convenience in use of the instruments available, the observations and calculations were made using metric units and the results were later translated into English units.

The data collected are for relative humidities of approximately 20, 45, 70 and 95 per cents, effective temperatures ranging from 44 to 100 F, and conditions of still air and velocities of 235 and 385 fpm. In fixing the limits of the effective temperature range, it was desired to obtain data over the range met within heating and ventilating practice. In order to establish the curves over this range,

[14] See Bibliography, p. 267.

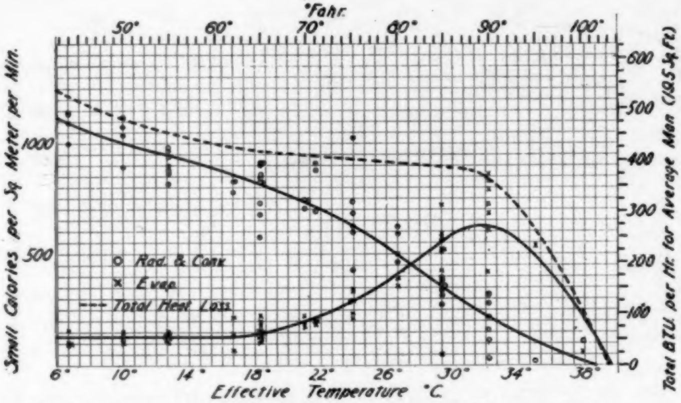

FIG 12. RELATION BETWEEN HEAT LOSS FROM THE HUMAN BODY AND EFFECTIVE TEMPERATURE AT 95% RELATIVE HUMIDITY

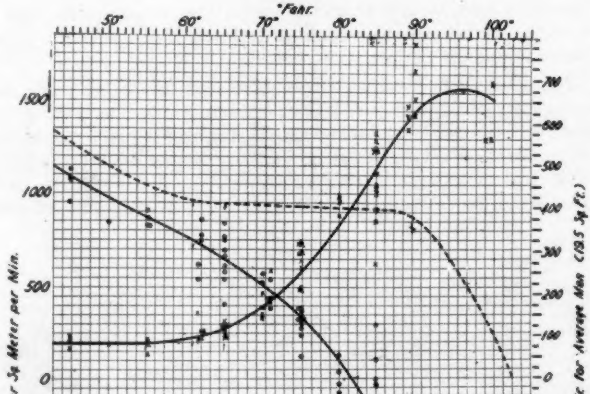

FIG. 13. RELATION BETWEEN HEAT LOSS FROM THE HUMAN BODY AND EFFECTIVE TEMPERATURE AT 20% RELATIVE HUMIDITY

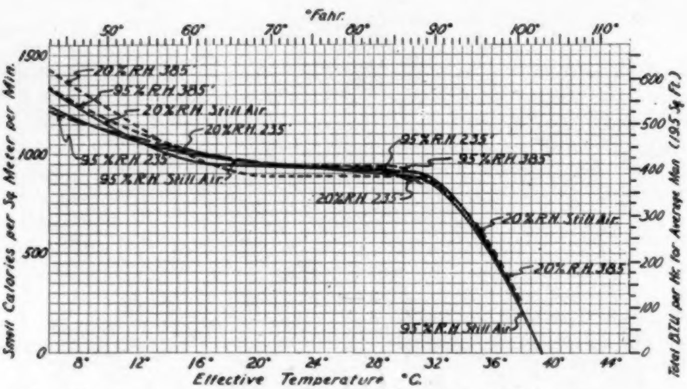

FIG. 14. RELATION BETWEEN HEAT LOSS FROM THE HUMAN BODY AND EFFECTIVE TEMPERATURE FOR STILL AND MOVING AIR

limits somewhat beyond those dictated by the needs of such practice were chosen. The upper practical limit was fixed at 90 F effective temperature. The lower limit was chosen as the lowest which could be endured at rest with the prescribed clothing without undue discomfort.

Beyond the limit of 90 F effective temperature four tests were made in order to check the tendency of the curves at higher temperatures. These points show a very definite direction for the curves but may be slightly in error due to the necessarily shorter duration of the test, the inability of the body to adjust itself quickly to abnormal conditions, the excessive temperature rises, the excessive perspiration and the general effect of the hot conditions on the subjects and observers.

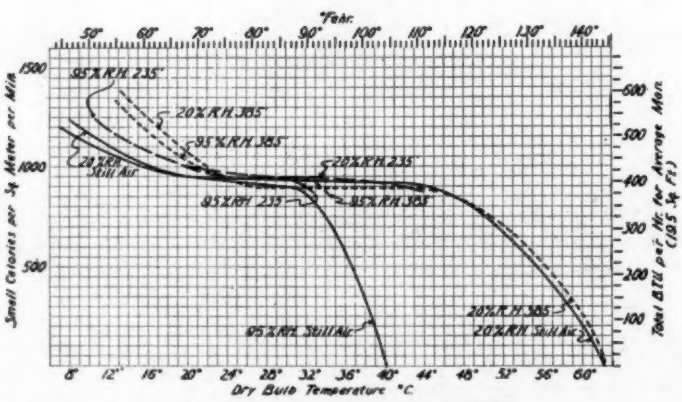

FIG. 15. RELATION BETWEEN HEAT LOSS FROM THE HUMAN BODY AND DRY-BULB TEMPERATURE FOR STILL AND MOVING AIR

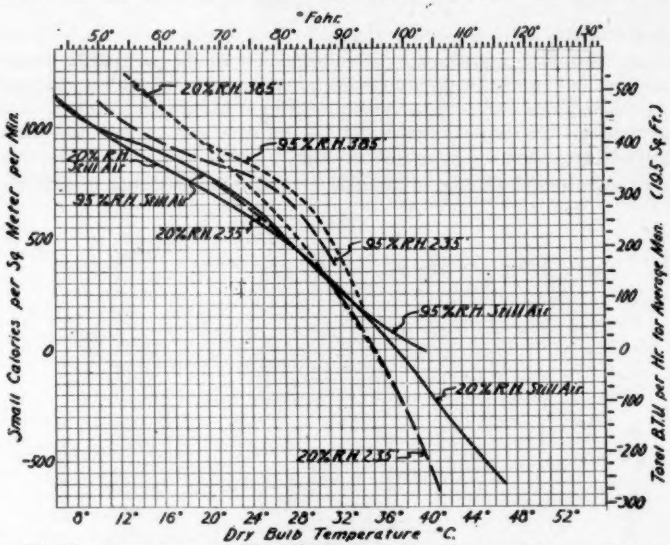

FIG. 16. RELATION BETWEEN HEAT LOSS FROM THE HUMAN BODY BY RADIATION AND CONVECTION AND DRY-BULB TEMPERATURE FOR STILL AND MOVING AIR

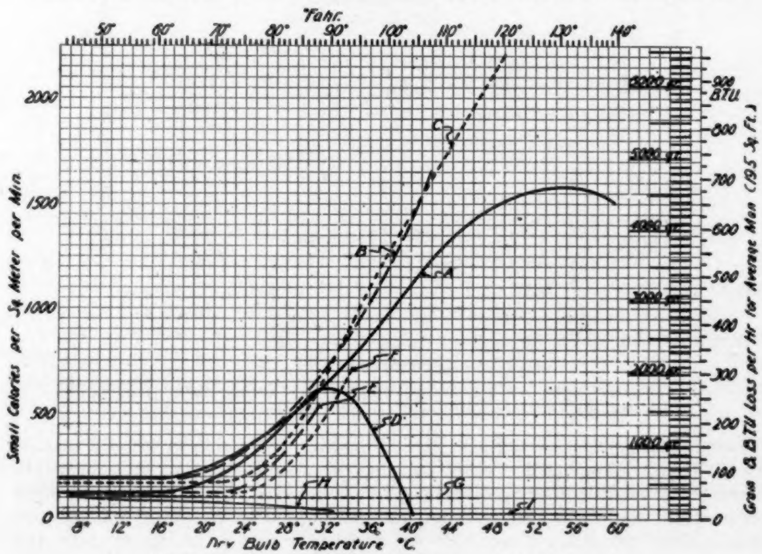

FIG. 17. HEAT AND WEIGHT LOSS BY EVAPORATION

A—From body surface and respiratory tract for 20% relative humidity and still air.
B—From body surface and respiratory tract for 20% relative humidity and 235 fpm vel.
C—From body surface and respiratory tract for 20% relative humidity and 385 fpm vel.
D—From body surface and respiratory tract for 95% relative humidity and still air.
E—From body surface and respiratory tract for 95% relative humidity and 235 fpm vel.
F—From body surface and respiratory tract for 95% relative humidity and 385 fpm vel.
G—From respiratory tract 20% relative humidity and still air.
H—From respiratory tract 95% relative humidity and still air.
I—Weight loss other than evaporation.

Data and Results

Fig. 6 gives the heat production in calories per minute per square meter of body surface, and also in Btu per hour for an average sized man, plotted against effective temperature. Individual symbols distinguish the tests at different relative humidities. Curves are shown for 20 per cent and 95 per cent relative humidity and also for the average of all tests.

It is a recognized physiological fact that all metabolic processes including res-

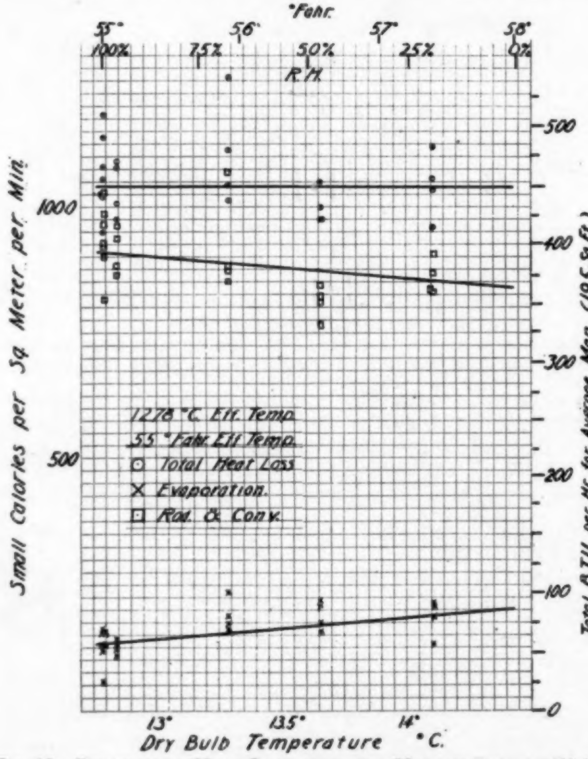

FIG. 18. RELATION OF HEAT LOSS FROM THE HUMAN BODY AT 55 F EFFECTIVE TEMPERATURE AND DRY-BULB TEMPERATURE

piration, heat production and heat dissipation are closely related to body surface area. This is accounted for by the fact that the main function of such metabolic processes is the control of production and dissipation of heat in order to maintain temperature equilibrium in varying atmospheric conditions. For man or any species of animal with a given degree of clothing or natural insulation (hair, fur or hide) heat loss and, therefore, metabolic processes are functions of the surface area of the body.

The heat production and loss values were first determined and plotted in units

of small calories per minute per square meter of body surface in accordance with accepted practice of the physiologist. For this purpose body surface area is taken from weight and height in the chart, Fig. 7, as given by Du Bois[15] and translated into English units. Heat production and loss in English units are given at the right hand side of the charts in Btu per hour for "a unit of surface area equal to that of an average-sized man," that is a 150 lb man, 5 ft 8 in. in height having a surface area of 19.5 sq ft or 1.81 sq m. The curves in Fig. 8 give the rate of heat loss per unit surface area and the total heat loss for each

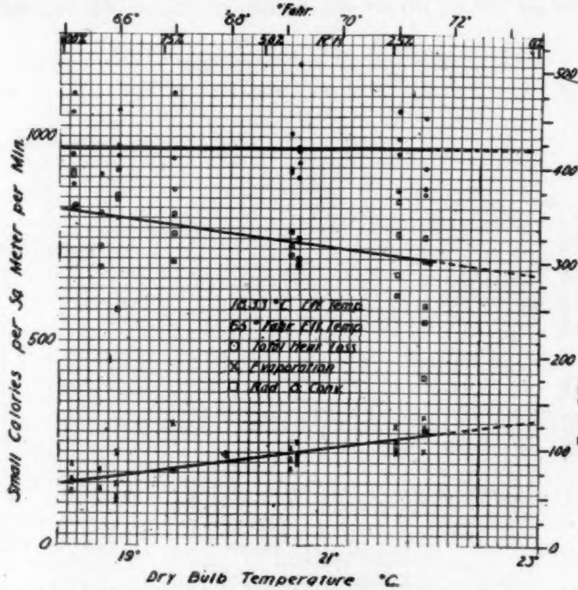

FIG. 19. RELATION OF HEAT LOSS FROM THE HUMAN BODY AT 65 F EFFECTIVE TEMPERATURE AND DRY-BULB TEMPERATURE

of the seven subjects entering the tests. There is much less variation in the curves for the seven men when heat loss per unit area of surface is plotted.

The total heat loss in calories per minute per square meter of body surface and also for a body surface equal to that of an average man is plotted against effective temperature for all the subjects at several relative humidities in still air in Fig. 9. Tests at different relative humidities are indicated by different symbols.

Heat loss by radiation and convection and by evaporation, respectively, are plotted against dry-bulb for all subjects in Figs. 10 and 11. The sensible and latent heat losses from the respiratory tract are also given. The latent and sensible losses from the respiratory tract were calculated from the weight, temperature and moisture content of the inspired air, and the temperature and degree of saturation of the exhaled breath as determined in this investigation.

Figs. 12 and 13 give composite pictures of total heat loss, and loss by radiation

[15] See Bibliography, p. 267.

and convection and by evaporation for two extremes of relative humidity. They also serve to demonstrate that heat loss by radiation and convection and by evaporation vary with relative humidity for the same effective temperature, or heat loss by radiation and convection and by evaporation are not functions of effective temperatures. Figs. 10 and 11 on the other hand show that within the range of practical application, heat loss by radiation and convection and by evaporation are functions of dry-bulb temperature.

Average curves giving total heat loss for all subjects, plotted against effective temperature for still air and for air velocities of 235 and 385 fpm, and for rela-

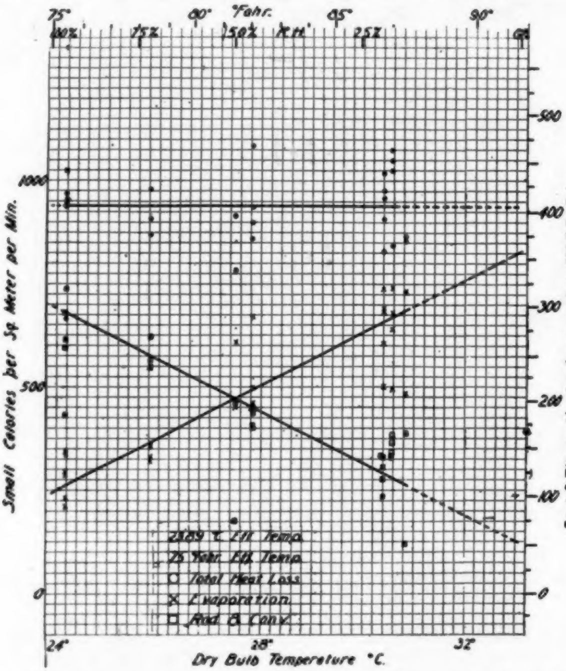

FIG. 20. RELATION OF HEAT LOSS FROM THE HUMAN BODY AT 75 F EFFECTIVE TEMPERATURE AND DRY-BULB TEMPERATURE

tive humidities of 20 and 95 per cent are presented in Fig. 14. The same curves are plotted in Fig. 15 against dry-bulb temperature. A comparison of the curves in these two figures demonstrates that total heat loss is a function of effective temperature for all relative humidities and for still and moving air rather than of dry-bulb temperature. The small divergence of the curves at low temperatures when plotted against effective temperature in Fig. 14 is no greater than might be expected, due to experimental error.

Total heat loss by radiation and convection and by evaporation, respectively, are given in Figs. 16 and 17, still air and air velocities of 235 and 385 fpm and for 20 per cent and 95 per cent relative humidity. Fig. 17 also gives the weight

loss by evaporation in grains per hour in a scale at the right-hand side of the chart. Curves are given for latent loss from the respiratory tract and weight loss from the entire body by other than evaporation.

Figs. 18, 19, 20 and 21 give the total loss and the loss by radiation and convection and by evaporation for varying dry-bulb temperatures along the 55, 65, 75 and 85 F effective temperature lines in still air. Besides giving a picture of variation of heat loss along the effective temperature lines chosen, these charts also serve to show the variation of dry-bulb temperature and relative humidity with constant effective temperature in different regions of the psychrometric chart.

A consideration of the heat production and total heat loss curves in Figs. 6, 9, 14 and 15 shows that heat production and loss are functions of effective tem-

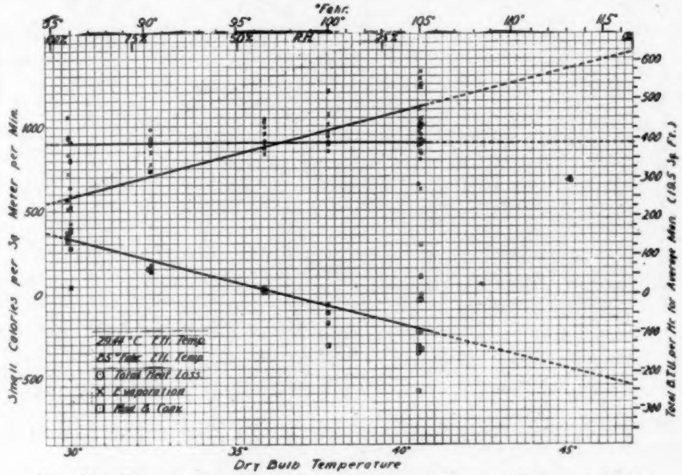

Fig. 21. Relation of Heat Loss from the Human Body at 85 F Effective Temperature and Dry-Bulb Temperature

perature for both still and moving air. Heat loss increases with fall in temperature below the comfort line, 65 F, as one would expect from the greater temperature difference between the body and air. The body compensates for this greater loss by an increase in heat production in order to maintain temperature equilibrium of the body at 98.6 F. In the tests here reported heat production does not entirely compensate for heat loss until a temperature of 80 F is reached with the result that there is a slight fall in body temperature which may, however, be due to decrease in muscular activity of the subjects upon entering the test rather than upon the test condition. From 65 F effective temperature to 87 F effective temperature heat production and loss are nearly constant. Above this temperature range heat production increases slightly while total heat loss falls off with increasing rate until it becomes zero at an effective temperature about 4 deg above normal body temperature.

Heat loss by radiation and convection is a function of dry-bulb temperature depending upon the difference between body and air temperature. As this differ-

ence decreases heat loss by radiation and convection decreases and becomes zero when the dry-bulb temperature of the air and surrounding objects and the temperature of the body are the same. For higher temperatures there is a reversal or heat is transferred from the air to the body. Air movement increases the heat exchange between the body and air by radiation and convection as measured. Actually, heat loss by radiation is not affected by air motion except in so far as skin temperature may be lowered. For either still or moving air heat loss by radiation and convection is largely in accordance with the physical laws of heat transfer and is little affected by physiological control.

If the true average temperature of the exposed surface of the clothed body were known it should be easy to formulate a law for the transfer of heat between the body and the air. Such a law would be expressed by an exponential equation

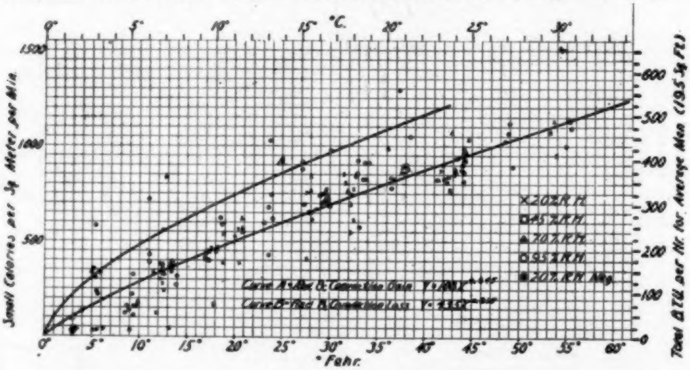

FIG. 22. RELATION BETWEEN HEAT LOSS FROM THE HUMAN BODY BY RADIATION AND CONVECTION AND THE DIFFERENCE BETWEEN BODY AND AIR TEMPERATURES

of the form $H = M\ (t_1 - t_2)^B$ or Log $H =$ Log $M + B$ Log $(t_1 - t_2)$
where $H =$ the rate of heat transfer per unit area.

M and $B =$ constants depending on conditions of body surface and surroundings.

$(t_1 - t_2) =$ temperature difference between body and air.

It is difficult, however, to evaluate the effective mean surface temperature of the clothed body because of variation of character of the surface exposed and the insulation or clothing worn. The only constant and fixed value is that of internal body temperature approximately 99 F. In Fig. 22 an attempt is made to fit an equation of the form given above to the data substituting (body temperature—air temperature) for $(t_1 - t_2)$. Curve A is for heat transfer from air to body for air temperatures above body temperature. Curve B is for normal heat loss from the body to the air.

Heat loss by evaporation although probably in accordance with the physical laws of evaporation is controlled to a great extent by physiological control of the availability of moisture for evaporation. Heat loss by evaporation is minimal for temperatures up to 65 or 73 F dry-bulb depending upon air motion. This is in the region where temperature equilibrium is maintained through control of heat production, to offset heat loss by radiation and convection. In the region

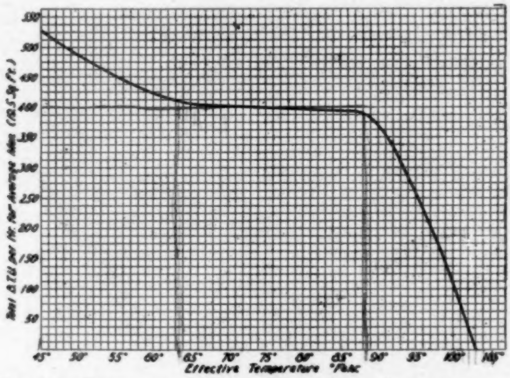

FIG. 23. RELATION BETWEEN TOTAL HEAT LOSS FROM THE HUMAN BODY AND EFFECTIVE TEMPERATURE

from 65 F to 87 F effective temperature (109.5 F dry-bulb with 20 per cent relative humidity and 88 F dry-bulb with 95 per cent relative humidity) where heat production is practically constant with a rapidly diminishing rate of heat loss by radiation and convection, the total rate of heat loss is maintained practically constant by control of evaporation through availability of moisture. From a minimum rate of heat loss by evaporation of (190 calories) 81.9 Btu per hour per average person at 55 F it increases to a maximum of (1570 calories) 676.7 Btu per hour at 96 F effective temperature and 20 per cent relative humidity.

Air motion reduces heat loss by evaporation when the dry-bulb temperature is 80 to 92 F dry-bulb depending on the relative humidity. For higher temperatures air motion increases heat loss by evaporation.

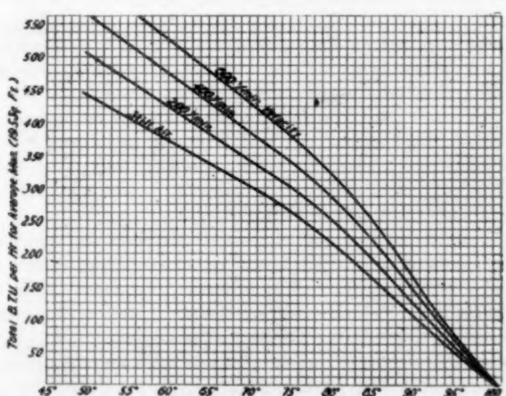

FIG. 24. RELATION BETWEEN SENSIBLE HEAT LOSS FROM HUMAN BODY AND DRY-BULB TEMPERATURE FOR STILL AND MOVING AIR

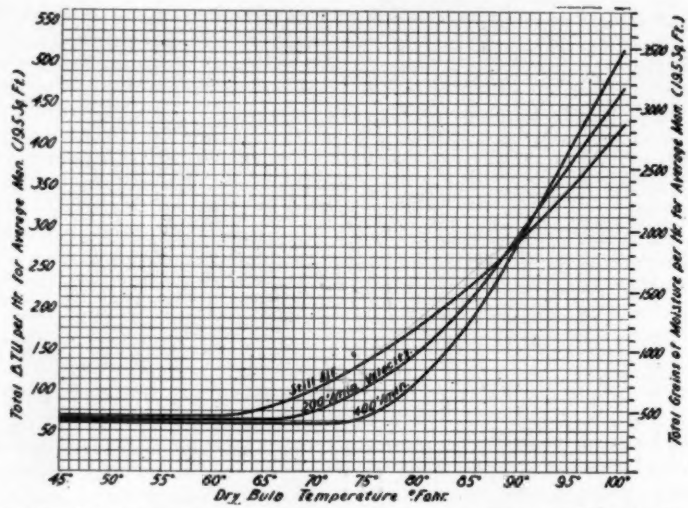

FIG. 25. RELATION BETWEEN HEAT AND WEIGHT LOSS FROM THE HUMAN BODY BY EVAPORATION AND DRY-BULB TEMPERATURE FOR STILL AND MOVING AIR

Practical Application of Heat and Moisture Loss Data to Air Conditioning Problems

The data given in Figs. 6 to 21 of this report are of practical value to the heating and ventilating engineer only in so far as they may be used in the solution of air conditioning problems. In the solution of such problems the engineer is interested in the total rate of heat dissipation to the atmosphere per average person and in the rate of sensible and latent heat dissipation, and also in the rate at which moisture is added to the atmosphere per person for all atmospheric conditions met with in practice. In the solution of certain problems it is also convenient to know the percentage of the total loss as sensible and latent heat. In Figs. 23, 24, 25 and 26 this information is given in convenient form for practical application. Although total heat loss and sensible and latent heat losses are not exact functions of effective and dry-bulb temperature, respectively, for all conditions of humidity and air motion, they are plotted as such in the curves. This is accomplished by approximations not always rigidly accurate but sufficiently so for application in most practical problems.

The sensible and latent heat loss curves for different air velocities in Figs. 24 and 25 are the result of approximations and extrapolations of the data contained in Figs. 6 to 21 and may not be rigidly accurate. However, since the total effect of air velocity on these losses is never very great, the approximation shown is probably sufficiently accurate for practical application.

With these precautions the data contained in Figs. 23 to 26 are offered for the use of the engineer in solving most of his practical problems. In some instances, however, in particular cases where extreme accuracy is desired, the variation in the data for different atmospheric conditions as given in Figs. 6 to 21 must be taken into consideration.

Typical uses of these data are given in the solution of the following problems:

Problem 1-A. How much sensible heat, how much latent heat and how much water vapor will be added per hour to the atmosphere of an auditorium by an audience of 1000 adults, when the dry- and wet-bulb temperatures are 75 F and 63.5 F, respectively?

B. If the dry- and wet-bulb temperature of the auditorium were 85 F and 63 F, respectively, how much heat and moisture would be dissipated to the atmosphere?

Solution. Problem 1-A. From Fig. 24 find the sensible heat loss per person for 75 F dry-bulb and still air to be 265 Btu per hour. From Fig. 25 find the latent heat loss per person for 75 F dry-bulb to be 134 Btu per hour and the moisture added to be 905 grains per hour. 1000 x 265 = 265,000 Btu sensible heat, 1000 x 134 = 134,000 Btu latent heat and 1000 x 905 = 905,000 grains or 129 lb of water vapor will be added per hour to the air in the auditorium.

These sensible and latent heat loss additions may also be found as follows: The effective temperature of the condition 75 F dry-bulb and 63.5 F wet-bulb is 70.3 F effective temperature. From Fig. 23 find 403 Btu as the total heat added to the air by a person for 70.3 F effective temperature. From Fig. 26 find the percentages of sensible and latent heat at 75 F dry-bulb to be 66.5 per cent and 33.5 per cent. The sensible heat added to the air in the auditorium is 1000 x 0.665 x 403 = 267,995 Btu per hour. The latent heat added is 1000 x 0.335 x 403 = 135,005 Btu per hour.

Solution. Problem 1-B. From Figs. 24 and 25, respectively, the sensible and latent heat losses per person for 85 F dry bulb are found to be 164 and 225 Btu per hour. The water vapor added to atmosphere is 1520 grains per hour. The audience will then add 164,000 Btu sensible heat, 225,000 Btu latent heat and 1,520,000 grains or 217 lb of water vapor to the air in the auditorium per hour.

Problem 2. Neglecting the gain or loss of heat to an auditorium by transmission or infiltration through the walls, windows and doors, how many cubic feet of outside air, with 65 F dry-bulb, 59 F wet-bulb and 63.1 F effective temperature must be added per hour to an auditorium containing 1000 people in order that the inside shall not exceed 75 and 65 F, respectively.

Solution. Problem 2. Figs. 24 and 25 give 265 Btu sensible heat and 905 grains of moisture as the additions per person with 75 F dry-bulb in the auditorium. Therefore 265,000 Btu of sensible heat and 905,000 grains of moisture will be added to the air in the auditorium per hour.

Taking 0.24 as the specific heat of air, 2.4 Btu per lb of air will be required to raise the dry-bulb temperature from 65 to 75 F and $\frac{265,000}{2.4}$ = 110,400 lb of air or 110,400

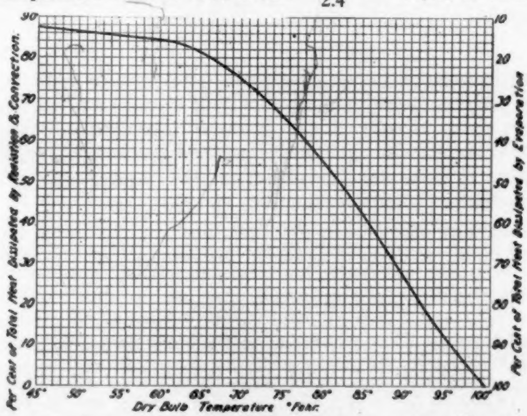

FIG. 26. RELATION BETWEEN HEAT LOSS FROM THE HUMAN BODY BY EVAPORATION, RADIATION, CONVECTION AND DRY-BULB TEMPERATURE

TABLE 2. CONDITION OF SENSIBLE PERSPIRATION FOR VARIOUS ATMOSPHERIC CONDITIONS

Degree of Perspiration*	Atmospheric Condition 95% R. H.			Atmospheric Condition 20% R. H.		
	E. T.	D. B.	W. B.	E. T.	D. B.	W. B.
Forehead clammy	**73.0**	73.6	72.4	**75.0**	87.0	60.7
Body clammy	**73.0**	73.6	72.4	**75.0**	87.0	60.7
Body damp	**79.0**	79.7	78.4	**81.0**	97.5	67.5
Beads on forehead	**80.0**	80.8	79.4	**87.0**	109.4	75.2
Body wet	**84.5**	85.4	84.0	**86.5**	108.5	74.6
Perspiration on forehead runs and drips	**88.0**	89.0	87.6	**94.0**	125.2	85.4
Perspiration runs down body	**88.5**	89.5	88.1	**90.0**	116.0	79.5

* 40 per cent of subjects registered degree of perspiration equal to or greater than indicated.

x 13.4 = 1,479,000 cu ft of air per hour will be required. This is equivalent to $\frac{1,479,000}{1000 \times 60} = 24.7$ cu ft per person per minute.

The moisture content of the inside air as taken from a psychrometric chart is 76 grains per lb of dry air and that of the outside condition is 65 grains so the increase in moisture content will be 11.0 grains per lb of dry air. $\frac{905,000}{11.0} = 82,300$ lb of dry air or approximately 83,000 lb of air at the specified condition will be required. This is equivalent to 83,000 x 13.4 = 1,112,000 cu ft of air per hour or $\frac{1,112,000}{1000 \times 60} = 18.5$ cu ft of air per minute per person.

The higher volume of 24.7 cu ft per person per minute will be required to keep the dry-bulb from rising above the 75 F specified. The wet-bulb will, therefore, not rise to the maximum of 65 F.

For good ventilation the condition of the air should never be such that sensible perspiration will result. In connection with the collection of the data contained in this report observations were made on the degree of noticeable perspiration experienced by the subjects for different atmospheric conditions. The lowest degree for which 40 per cent of the subjects experienced the conditions of perspiration indicated is given in Table 2 for 20 and 95 per cent relative humidity.

Summary

This report contains data on total heat loss, heat loss by radiation and convection and by evaporation per unit area of body surface and also for the surface area of a person of average size for various temperatures, humidities and air velocities. It also contains data on the moisture loss and degree of perspiration.

The relation of these losses to dry-bulb temperature, relative humidity and effective temperature is shown by curves and discussion.

Charts containing the data in practical form and examples showing how it may be used in the solution of practical problems in air conditioning are presented.

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DISCUSSION

PRESIDENT WILLARD: You have just listened to a most remarkable presentation. In the last thirty minutes, Director Houghten, to my mind, has summarized the basic principles of the new science of air conditioning. He has brought up to the minute the results of many years of painstaking research, research conducted jointly by the U. S. Bureau of Mines, the U. S. Public Health Service, several universities, and this Society.

I said in an earlier session that we are moving into a zone of complicated environment concerning our knowledge of the many factors affecting the practice of the heating and ventilating engineer. You have just seen a very complicated section uncovered. Are we all going to be specialists? Have we got to become specialists?

I sit here and hear the Research Staff present these advanced papers—and they are advanced papers—and I am wondering how many men in the Society are in a position to realize and understand just the significance of these results, and where they are leading us, where we are going. We may not like it. We may not enthuse over the complications that confront us in all lines of engineering work, and in all lines of professional knowledge even outside of engineering, but they are there; we are spending money to set up these machines, these staffs, these agencies to secure more knowledge, and we are getting it; we are getting it in such shape that the men who study and the men who understand, can use it, and those men are going to surpass and are going to be successful in the advancement of the science of heating and ventilation. I am using the word science because it seems to me that finally we have something in the field of air conditioning which is on a scientific basis. I am amazed at the long distance we have come in a relatively few years, as a result of the concerted efforts of this Society, to first establish, then maintain, and continue the work of a trained research staff under a full time director who devotes his entire thought and energy to the problem.

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That is the only way we will make progress, by concerted effort on a definite program, with a director who is giving his full time and attention to the problem in hand. It seems to me that the Research Staff should be congratulated on what it has accomplished.

G. L. Larson: I feel that this is one of the most valuable papers that has ever been presented before this Society. There is one question that I would like to ask Mr. Houghten, and that is, have any studies been made to find out the heat dissipation of persons doing violent exercise?

We have had a recent experience up at Madison that bears directly on this point. During the last year we built a new Memorial Union and the banquet hall in that building during the past semester has been used for dancing purposes. The amount of ventilation supplied complies with the state code of Wisconsin, but recently we have been receiving complaints that the ventilation was entirely inadequate.

About a week or so ago we made some tests to find out what was taking place. On this particular night it was —5 outside during the entire evening, and by means of a supercharger or possibly a shoe horn they had managed to get about 1,000 people into that dance hall, although it has only 5,000 sq ft of dancing surface. Now, I asked the question about violent exercise because for some of us dancing is violent exercise. We installed in the outgoing vent, that is, the foul air vent, a recording thermometer and also measured the humidity and the results were rather interesting. Although the ventilating system was operating and supplying more air than required by the state ventilation code the recording thermometer showed that the temperature of the outgoing air was going up about 2 deg per dance. At the end of the third dance, the humidity had gone from 18 to 32 per cent, and before the night was over, we were blowing into that room air at 39 F and still we could not keep the room temperature to 70 F.

So there is a problem that I am very much interested in, and therefore, I want to ask the question, have any studies been made that would show the amount of heat dissipated under those conditions?

Now, you wonder how people can dance under those conditions. Well, if you have been at any of the state universities lately, you will know when you see a dance like that going on, it is like starting a molded piece of jelly in motion, the top moves but the bottom does not.

F. C. Houghten: No study has been made at the Laboratory of the increase in heat loss from the body with work. It is the intention of the Committee that this should be studied as a part of the future laboratory program. It is well known that for a person working violently or walking at a high rate, the rate of heat production increases materially, perhaps as much as ten times. However, nothing is known at this time as to whether this increase in heat is lost by radiation and convection or by evaporation, except as we can surmise from the results reported today. Probably as more heat is produced and necessarily lost, radiation and convection loss is not much affected, but evaporation loss increases so as to account for the increase in the total.

Professor Larson: I might say these were rough calculations. We did not know what rate to use as the heat given off per person. We used about 550 Btu's; on this basis the heat given off by the crowd was over twice the heat loss of that room when it was —5 F outside.

No. 831

THE SUMMER COMFORT ZONE: CLIMATE AND CLOTHING[1]

By C. P. YAGLOU[2] AND PHILIP DRINKER,[3] BOSTON, MASS.
MEMBERS

INTRODUCTION

IN a previous paper (1), it was shown that standards of comfort, in so far as temperature, humidity and air movement are concerned, are not absolute, but are considerably affected by climatic and other conditions. This variation appears to be caused by differences in the clothing worn in different climates and at different times of year, as well as by the adaptive physiologic changes[4] which take place in man in response to climatic changes.

The work here described deals with the summer comfort zone as determined by experiments on a large group of men and women wearing customary warm weather clothing and engaged in sedentary pursuits. It includes also a study of the relation of climate and season to the type of underclothing worn.

EXPERIMENTAL PROCEDURE

The experiments were carried out in the psychrometric chamber of the Harvard School of Public Health (2). This room is equipped with a complete heating, ventilating and air conditioning system for maintaining the desired temperature, and humidity conditions.

Preliminary work on the summer comfort zone was begun in July, 1926, but the greater part of the work was done during July, August and September of 1927. July and August were cooler in 1927 than in other years and September was warmer.

The comfort zone was determined as a result of fifteen experiments, eight of which were conducted with progressively increasing room temperatures, (usually in steps of 2 F or 3 F at a time) and the other seven with decreasing temperatures, in order that the probable optimum temperature might be approached from both the cool and the warm side of the zone. In this way, adaptation to diurnal changes in air conditions was taken into account. In the course of the fifteen tests, a total of 2901 votes on sensations of comfort were secured.

In addition to these fifteen tests, four special experiments were undertaken for the purpose of determining the influence of room occupancy on the optimum temperature. These data are treated separately.

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[1] From the Department of Ventilation and Illumination, Harvard School of Public Health, Boston, Mass.

[2] Instructor in Ventilating and Illumination, Harvard School of Public Health, Boston, Mass.

[3] Assistant Professor of Ventilation and Illumination, Harvard School of Public Health, Boston, Mass.

[4] This is usually referred to in the literature as "changes in acclimatization."

Presented at the Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, January, 1929.

TABLE 1. COMFORT VOTES CLASSIFIED ACCORDING TO EFFECTIVE TEMPERATURE

Sensations of Comfort	Effective Temperature (F) (Normal Scale)																
	62–63	63–64	64–65	65–66	66–67	67–68	68–69	69–70	70–71	71–72	72–73	73–74	74–75	75–76	76–77	77–78	78–79
Increasing Temperature																	
Number of Votes Recorded																	
1. Cold	..	..	78	64	61	27	11	11	2	..	..	..	..	..	..	..	..
2. Comfortably cool	..	..	2	14	54	50	57	37	33	10	0	1	..	..	..	..	..
3. Very comfortable	..	..	..	..	13	27	53	108	133	56	48	44	14	8	2	..	..
4. Comfortably warm	..	..	..	..	..	..	4	2	16	35	42	45	34	22	9	2	1
5. Too warm	..	..	..	..	..	..	..	..	3	2	6	10	42	28	50	53	59
Totals	..	..	80	78	128	104	125	158	187	103	96	100	90	58	61	55	60
Decreasing Temperature																	
Number of Votes Recorded																	
1. Cold	9	80	75	71	45	29	24	17	2	1	..	..	..	..	..	..	..
2. Comfortably cool	..	..	3	18	22	44	50	38	32	19	4	3	..	..	..	..	..
3. Very comfortable	..	..	..	1	5	14	25	37	77	68	67	43	27	7	..	..	..
4. Comfortably warm	..	..	..	..	..	..	..	..	6	23	49	51	58	24	10	9	..
5. Too warm	..	..	..	..	..	..	..	..	..	5	25	35	56	36	32	42	..
Totals	9	80	78	90	72	87	99	92	117	116	145	132	141	67	42	51	..

We have adopted the same classification for sensations of comfort that was used in a previous study (1), *viz.*:

1. Cold
2. Comfortably cool
3. Very comfortable
4. Comfortably warm
5. Too warm.

Altogether, ninety-one subjects took part in these experiments—fifty-six men and thirty-five women. About half of them were summer school students or employees of the Harvard Medical School; the other half were secured from employment agencies in Boston. The ages of the men varied from 22 to 57 years and those of the women from 22 to 72 years; in both instances the majority were between 22 and 37 years of age. No restrictions were imposed in regard to clothing, the subjects being requested to dress in their usual way. The majority of the men wore two-piece palm beach or light weight woolen suits and the women silk, linen or cotton dresses.

Most of the experiments took place in the afternoon between 1:50 and 5:30 o'clock.

The procedure followed was approximately the same as that used in our previous work (1), with the following exceptions:

(*a*) It was found necessary in some experiments to increase the preliminary period allowed for adaptation from one-half to one and one-half hours, in order to overcome the influence of the preceding environment.

(*b*) No special pains were taken to maintain definite percentages of relative humidity during the experiments, as it was found in previous studies (1), (3) that the sensations of comfort followed the effective temperature[5] scale closely, regardless of humidity. The relative humidity in the present experiments varied from day to day between the approximate limits of 45 and 75 per cent.

(*c*) Dry and wet kata-thermometer readings were taken in all experiments. These data, however, will be given in another paper.

(*d*) Outdoor temperatures were taken on the days the tests were made, for the purpose of determining the influence of outdoor temperature on the optimum temperature[6] indoors.

Air movement in the psychrometric room was determined from the cooling powers of the dry kata-thermometer. By regulating the speed of the supply and exhaust fans and by manipulating dampers, the velocity of the ventilating current was kept between the limits of 15 and 25 fpm, as was done in the Pittsburgh experiments on the winter comfort zone (3).

DATA AND DISCUSSION OF RESULTS

The comfort votes obtained in the fifteen regular experiments are classified in Table 1 according to effective temperature. The upper half of the table gives the number of votes for each degree of effective temperature recorded in those ex-

[5] Effective temperature is an index of the sensations of warmth or cold felt in response to temperature, humidity and movement of the air. When the dry- and wet-bulb temperatures and the rate of air movement are known, the effective temperature can be computed from charts or tables. This index was determined experimentally at the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. It comprises two different scales: the *basic* (4) for men stripped to the waist and the *normal* (5) for men normally clothed.

[6] The term *optimum temperature* is used in this paper to signify the most comfortable temperature and should be understood in this restricted sense. Since the prolonged effects of temperature, humidity and air movement on health are not so well known as their effects on comfort, the optimum conditions for comfort cannot be regarded as necessarily identical with those for health.

periments in which the room temperature was progressively increased; the lower half of the table gives the data obtained with progressively decreasing room temperatures.

Sensations of Comfort in Relation to Effective Temperature

In Figs. 1 and 2, the comfort votes are plotted in percentages as ordinates against effective temperature as abscissas, for increasing and decreasing room

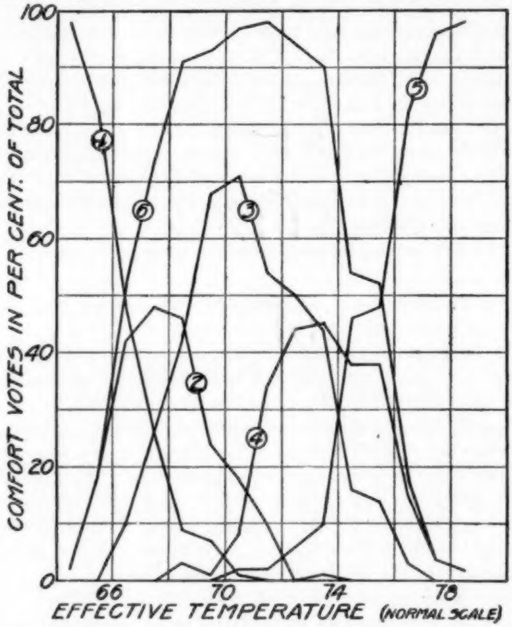

FIG. 1. SENSATIONS OF COMFORT IN RELATION TO EFFECTIVE TEMPERATURE. OBSERVATIONS WITH INCREASING ROOM TEMPERATURE
(1 = cold; 2 = comfortably cool; 3 = very comfortable; 4 = comfortably warm; 5 = too warm; 6 = sum of 2, 3 and 4.)

temperatures, respectively. The graphs in these two figures are similar and show an optimum temperature of 70.5 F effective temperature, a cold limit of 63.5 F to 64.5 F effective temperature, and a warm limit of 78.5 F to 79.5 F effective temperature. The coincidence of these values in Figs. 1 and 2 is significant, because it indicates that the diurnal adaptation to changing air conditions has been sufficiently allowed for by our experimental method.

Fig. 3 shows the combined data of Figs. 1 and 2, from which the limits of the comfort zone and the optimum temperature for summer conditions can be taken, as follows:

	Effective Temperature
Probable cold limit	64.0 F
Probable optimum	70.5 F
Probable warm limit	79.0 F

The curves in Figs. 1, 2 and 3 are of the asymmetric frequency type and, on that

account, the optimum temperature does not fall in the middle of the zone but is nearer to the cold limit. This observation is thoroughly in accord with the results of previous work in determining comfort zones (1, p. 257), (3, p. 373). The skewness of the curves may possibly be attributed to the fact that, owing to our custom of protecting the body by clothing and artificial heat, the chemical regulation of body heat (*i. e.* by increase in metabolism) is less well developed

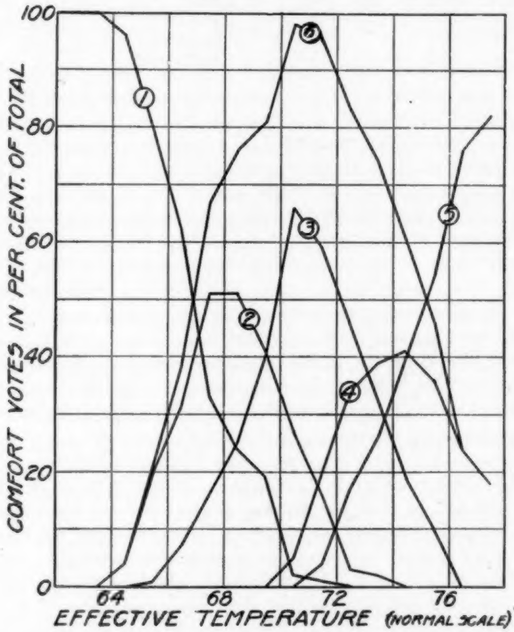

FIG. 2. SENSATIONS OF COMFORT IN RELATION TO EFFECTIVE TEMPERATURE. OBSERVATIONS WITH DECREASING ROOM TEMPERATURE
(1, 2, etc., as in Fig. 1.)

than the physical regulation (*i. e.* by sweating); in consequence we are better capable of perceiving sensations of cold than of heat.

Diurnal and Monthly Changes in Outdoor Temperature in Relation to Optimum Indoor Temperature

In order to make crowded places comfortable during warm summer weather, American ventilating engineers have adopted the practice of maintaining indoors, by means of refrigeration, a dry-bulb temperature 8 deg to 15 F lower than that prevailing out of doors. Their chief purpose is not so much to reduce the temperature to the optimum degree as to maintain a reasonably comfortable temperature and, at the same time, avoid causing sensations of chill or of intense heat in entering and leaving the building.

This practice is justified in many crowded public buildings where the patrons spend a comparatively short period of time. According to our experiments, how-

TABLE 2. DIURNAL AND AVERAGE MONTHLY OUTDOOR TEMPERATURE IN RELATION TO OPTIMUM INDOOR TEMPERATURE

Prevailing Outdoor Dry-Bulb Temperature	Probable Optimum Effective Temperature (Indoors)	Number of Observations	Month	Monthly Average Outdoor Dry-Bulb Temperature (Boston, Mass. 1927)
F	F			F
99-90	70.2	84	July	70.5
89-80	70.9	91	August	67.4
79-70	70.4	109	September	65.0
below 70	69.3	15	September	65.0

ever, it is not adequate for indoor exposures of one and one-half hours or more —for offices, residences and the like. This may be seen from Table 2, where the optimum temperature is classified in four groups, according to the out-of-door temperature. There seems to be no significant relationship between the two, since the optimum temperature in July, August and September remained practically the same, in spite of the fact that the outdoor temperature actually varied from 99.5 to 70 F. When the outdoor temperature was below 70 F when 66 F was registered—the optimum temperature did drop slightly, as shown in the table, but too much stress must not be laid upon this circumstance because of the limited number of observations, all made in the course of a single experiment. Nevertheless, it is clear that 70 F or slightly less, is the lowest summer or winter temperature to which people in the United States are adapted in their living and working quarters. It may be taken as the temperature above and below which significant changes in adaptation and clothing are to be expected.

A much more striking correspondence becomes apparent in comparing changes in optimum temperature with changes in the average monthly outdoor temperature. It will be seen from Table 2 that the latter varied by only 5.5 F from July to September, and it is probably for this reason that the optimum temperature changed so little. The human body cannot, as we have shown elsewhere (1, p. 257), adapt itself readily to changing temperature conditions, but it probably follows the law of averages. The transitory changes in outdoor temperature are only partially felt indoors, because they are mitigated to a considerable extent by passage through buildings. The result is that we become adapted to an indoor temperature which is more equable than that out of doors and which, during the summer season, runs parallel with the mean outdoor temperature. Diurnal changes in outdoor temperature, therefore, may be assumed to affect the optimum indoor temperature, in so far as they modify the mean seasonal temperature.

Badham and others (6, p. 57), in a study of atmospheric conditions in theaters in Sydney, New South Wales, Australia, also conclude that diurnal differences between indoor and outdoor temperature have no apparent influence on sensations of comfort, so long as the other physical conditions remain the same.

Seasonal Variation in Comfort Zone

The influence of season on the comfort zone can be observed from Table 3, where the summer zone limits are compared with those found in winter for subjects wearing customary indoor winter clothing. Before proceeding with the discussion, it should be made clear that the zone limits given in the table include the entire range of air conditions voted as cold or warm by all the subjects. The data on the winter zone were obtained from the original report (3, p. 371, Fig.

TABLE 3. SEASONAL VARIATION IN COMFORT ZONE

Comfort Zone	Effective Temperature (F) (Normal Scale)				Dry-Bulb Temperature (F) (Relative Humidity 50 Per Cent)			
	Probable Cold Limit	Probable Optimum	Probable Warm Limit	Width of Zone	Probable Cold Limit	Probable Optimum	Probable Warm Limit	Width of Zone
Summer	64.0	70.5	79.0	15.0	67.5	75.7	86.8	19.3
Winter	60.0	66.0	74.0	14.0	62.6	70.0	80.0	17.4
Difference	4.0	4.5	5.0	1.0	4.9	5.7	6.8	1.9

9) and the effective temperature was converted from the basic to the normal scale. Table 3 shows that the summer zone is 1 deg effective temperature wider than the winter zone and that it has shifted to higher temperatures. The probable cold limit rose from 60 F in winter to 64 F effective temperature in summer, the probable optimum from 66 F to 70.5 F effective temperature, and the probable warm limit from 74 F to 79 F effective temperature. Corresponding increases in dry-bulb temperature, when the relative humidity is constant at 50 per cent, are shown in the right half of the table. We ascribe this variation partly to adaptation to seasonal weather and partly to differences in the clothing worn in the two seasons, as we stated previously.

Vernon (7, p. 394) observed, by means of the kata-thermometer, similar seasonal changes in British factories. He found that, in order to produce a given sensation of air movement, it was necessary to have a dry kata cooling power of about 1 millicalorie per square centimeter per second higher in winter than in summer. This difference he attributes chiefly to acclimatization.

Badham and others (6, pp. 59 and 56) who used Vernon's method in their observations in the Sydney theaters, found little difference between the kata cooling power required to produce a given sensation in winter and that required in summer, but they did observe considerable seasonal change in sensations of air movement. Their results were secured chiefly from the subjective sensations of two investigators, and this was also true of Vernon's experiments.

Influence of Room Occupancy on Optimum Temperature

The greater part of our experiments were carried out with about fourteen people in the test chamber. The gross floor area per occupant was 25 sq ft and there was an air space of about 220 cu ft per person. Allowing for aisles and equipment, the net floor area per occupant was reduced to 11.5 sq ft. This condition is comparable to that found in uncrowded lecture and recitation rooms. When the number of occupants was reduced to eight, there was no appreciable difference in the results; when the number exceeded twenty, however, the comfort votes were noticeably affected.

In order to obtain a quantitative measure of this effect, we conducted four special experiments, two with eight subjects in the psychrometric room and the other two with the same eight subjects and seventeen others in addition, so that, with a total of twenty-five persons, the room was reasonably crowded. In all four tests, the room temperature was adjusted to the point of maximum comfort for the original eight subjects, according to their votes, and the relative humidity was kept at about 60 per cent.

Table 4 gives the average results. The probable optimum effective temperature for the eight subjects was 70.8 F when the gross floor area per occupant was

44 sq ft and the air space per occupant 380 cu ft; it was 69.4 F when the floor area was 14 sq ft and the air space 120 cu ft per person. The difference is about 1.5 F in both dry-bulb and effective temperatures. The cause of this difference is obvious. In the more crowded room, the heat loss by radiation from the bodies of the subjects to the walls and other surroundings was reduced on account of counter radiation between subjects in close proximity. A lower air temperature

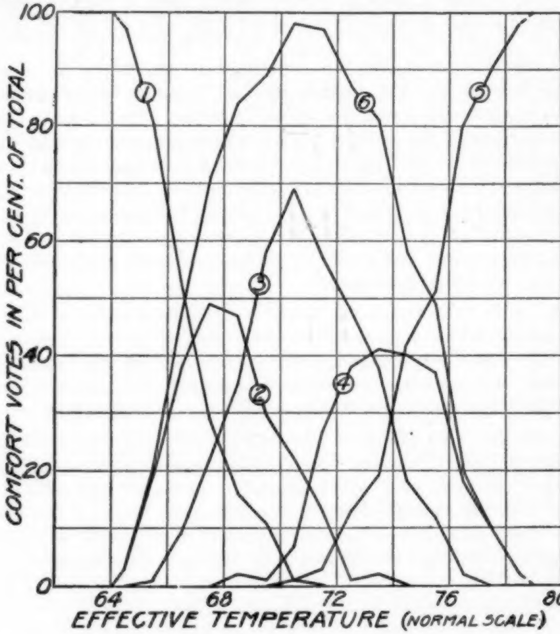

FIG. 3. SENSATIONS OF COMFORT IN RELATION TO EFFECTIVE TEMPERATURE. SUMMATION OF DATA IN FIGS. 1 AND 2 (1, 2, etc., as in Fig. 1.)

was therefore required for comfort, in order to offset the decrease in the heat loss by radiation through an increase in the heat loss by convection.

According to these data, one would expect to find a lower optimum temperature in auditoriums and schoolrooms than in residences, offices and the like. Although this is probably true for the winter season, in summer an additional factor must be considered. People entering a room from the outside in warm weather are often in an overheated condition and the evaporation of moisture from their damp clothing may produce a cooling effect which more than offsets the decrease in optimum temperature due to crowding.

Comfort Chart

Fig. 4 shows the effective temperature chart developed at the Pittsburgh laboratory (3, p. 372, Fig. 10) with the summer and winter comfort zones superimposed upon it. The variation in the sensations of comfort within this

zone is indicated by comfort scales. The scale values for the summer comfort zone were obtained from Graph 6, Fig. 3, of this paper and those for the winter zone from Graph 6, Fig. 9, of the Pittsburgh report (3, p. 371), after converting the scale of effective temperature from the basic to the normal.

Given the dry- and the wet-bulb temperatures, the effective temperature is found and the probable comfort of the condition determined by referring to the proper comfort scale. This chart is applicable for air movements between 15 and 25 fpm. Unfortunately, we have as yet no accurate data on the thermoequivalent conditions of temperature, humidity and air movement for velocities between 25 and 150 fpm. Although such data have, in the past, been estimated by interpolation from observations which actually covered velocities of 15 to 25, 150, 300 and 500 fpm (5), recent unpublished observations made in this laboratory indicate that this interpolation is inaccurate and that the region of air velocities between 20 and 150 fpm needs special study.

TABLE 4. INFLUENCE OF ROOM OCCUPANCY ON OPTIMUM TEMPERATURE

Number of Subjects in Test Chamber	Floor Area per Occupant		Air Space per Occupant	Most Comfortable Dry-Bulb Temperature	Relative Humidity	Probable Optimum Effective Temperature	Outdoor Temperature	Number of Observations (8 Control Subjects Only)
	Gross	Net						
	Sq Ft	Sq Ft	Cu Ft	F	Per Cent	F	F	
8	44	20.5	380	75.0	60	70.8	77.0	179
25	14	7.5	120	73.4	59	69.4	81.9	164

These comfort zones are not applicable where climate, clothing and heating standards differ substantially from those under which the zones were determined.

CLIMATE AND CLOTHING

A discussion of optimum temperature conditions would be incomplete without considering the factor of clothing, since this, too, is primarily concerned with the regulation of body heat. Climate, of course, largely determines the kind and amount of clothing worn, but its effects are modified to some extent by artificial control of the indoor atmosphere and by national customs and fashions. Differences in international comfort standards may be attributed in part to differences in these two factors—climate and clothing.

In order to study the relationship between climate and clothing, we secured data on the types of underclothing sold in the cold and warm seasons throughout the populous sections of the United States. In order to increase the number of observations for cold weather, we included in our list a few Canadian cities also. This information was secured from eighty men's clothing stores in thirty different cities (Table 5). The cold season was considered to extend from November 1 to May 1 and the warm season from May 1 to November 1. The types of underwear were classified as follows: (*a*) pure wool, (*b*) wool and cotton, (*c*) medium weight cotton, and (*d*) light-weight cotton. Table 5 also gives the seasonal outdoor temperature, humidity and wind velocity for each city as averaged from the records of the United States Weather Bureau. The temperatures given are the averages for the last fifty-five years. The humidity and wind velocity figures are the averages for the year 1925 alone, but are probably a fair representation.

In Fig. 5, the data from Table 5 are plotted in four separate graphs, showing the percentage of the four kinds of underwear sold in the different localities according to their seasonal outdoor temperatures. Although comparatively little woolen underwear is sold in the United States even in cold weather, a definite relationship appears to exist between the mean outdoor temperature and the kind

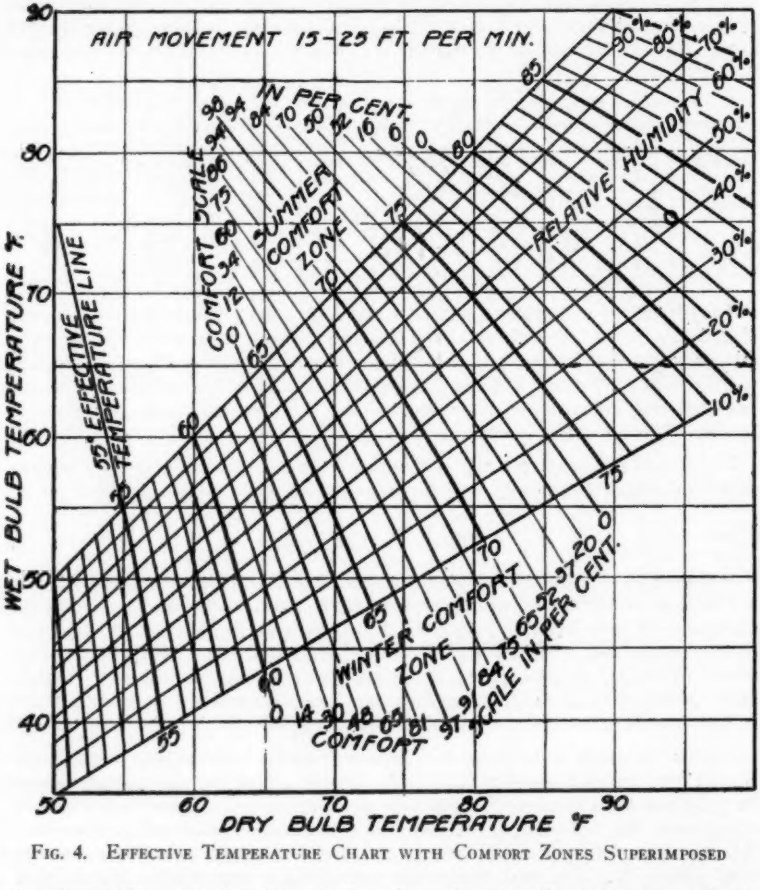

FIG. 4. EFFECTIVE TEMPERATURE CHART WITH COMFORT ZONES SUPERIMPOSED

of underclothing worn by the majority of people, regardless of season. Thus, pure wool is popular only in localities where the average winter temperature falls below 20 F; in cities where the winter temperature varies between 20 F and 37 F, wool and cotton mixtures are worn by the majority; in warmer cities, where the winter temperature is from 38 F to 50 F, medium weight cotton is widely used; when the average temperature exceeds 50 F, whether in summer or in winter, the greater part of the underwear sold is of the light weight cotton variety.

To take the city of Boston as an example, the percentages of the different weights of underwear sold there in the cool and warm seasons are as follows:

Season	Pure Wool	Wool and Cotton Mixtures	Medium Weight Cotton	Light Weight Cotton
	Per Cent	Per Cent	Per Cent	Per Cent
Cool (35.5 F)	10	40	33	17
Warm (63.7 F)	2	5	18	75

These values have been taken from Fig. 5 and agree as closely as can be expected with those given in Table 5. According to these curves, the representative kinds of underwear sold in Boston are wool and cotton mixtures in the cold season and light weight cotton in the warm season.

In Canada, to judge from the three Canadian cities included in Table 5, it appears that there is an appreciably greater percentage of woolen underwear purchased in summer than in cities in the United States where the same or lower summer temperatures prevail.

All four curves in Fig. 5 are of the same general type; they are asymptotic to the temperature axis, indicating that some wool and some light-weight cottons are worn at all seasons, no matter what the temperature may be. All four curves show fairly well defined model regions which represent, approximately, the most popular types of underwear for the specific climates.

The average humidity and wind velocity of the localities included in our study seem to have no noticeable influence on the type of underclothing worn.

The seasonal changes in clothing are not, of course, confined to underclothing. We obtained no data on outer clothing from the clothing stores, but, to judge from the practice of our experimental subjects, the seasonal changes in outer clothing are at least as significant as changes in underwear. Thus the greater number of our men subjects wore two-piece palm beach or light weight woolen suits in summer and medium weight woolen or woolen mixtures in winter.

Several of the firms submitting data were of the opinion that the increasing use of the closed and heated automobile and the better heating facilities in living and working quarters accounted for the reduction in the amount of woolen underwear sold today, as compared with that sold ten or twenty years ago.

Influence of Climate, Clothing and Heating Methods upon Comfort Standards

A comparison between the American seasonal optimum temperatures and those found by Vernon in England (7, p. 395, Table 2) shows that the American people prefer, in winter and in summer, indoor dry-bulb temperatures (Vernon gives no wet-bulb readings) which are about 8 F higher than those considered comfortable in England. In this comparison, we have assumed that Vernon's two investigators, on whose sensations his data rested, are representative of the average sedentary person in England and that Vernon's sensations of medium air movement are comparable to our sensations of comfort.

We ascribe this difference in comfort standards in part to the differences in clothing worn in the two countries. As we have pointed out, little woolen underwear is used in the United States even in cold weather; in England, we believe the contrary is true: woolen underclothing is commonly worn in winter and some prefer it in summer also, in spite of the fact that the English climate is, in

TABLE 5. SEASONAL VARIATION IN UNDERCLOTHING WORN BY MEN IN THE UNITED STATES

City and State	Cool Season, November to May							Warm Season, May to November						
	Average Weather Conditions			Kind of Underwear Worn				Average Weather Conditions			Kind of Underwear Worn			
	Dry-Bulb Temperature	Relative Humidity	Wind Velocity	Pure Wool	Wool and Cotton Mixtures	Medium Weight Cotton	Light Weight Cotton	Dry-Bulb Temperature	Relative Humidity	Wind Velocity	Pure Wool	Wool and Cotton Mixtures	Medium Weight Cotton	Light Weight Cotton
	F	%	miles/hr.	%	%	%	%	F	%	miles/hr.	%	%	%	%
Atlanta, Ga.	51.3	67	12.8	7	17	28	48	72.7	67	9.0	0	0	6	94
Bangor, Me.	29.7	70	11.5	60	30	10	0	57.2	72	8.9	0	0	20	80
Boston, Mass.	35.5	62	11.5	1	35	56	8	63.7	66	9.2	0	2	35	63
Buffalo, N. Y.	32.0	77	20.3	10	39	16	35	62.0	76	14.7	1	11	22	66
Butte, Mont.	28.6	65	7.6	20	45	29	6	58.3	47	8.4	1	5	9	85
Chicago, Ill.	34.6	68	12.5	2	31	36	31	65.8	65	10.3	0	1	2	97
Dallas, Tex.	51.8	65	8.9	2	6	47	45	78.0	61	8.5	0	1	14	85
Denver, Colo.	36.8	55	7.6	5	28	39	28	63.3	43	7.2	0	2	12	86
Detroit, Mich.	33.0	73	12.7	5	38	29	28	64.0	62	9.3	1	3	11	85
Duluth, Minn.	29.0	81	14.3	5	50	35	10	55.0	73	11.7	2	6	25	67
El Paso, Tex.	51.8	38	10.6	2	15	58	25	74.8	33	10.4	0	0	10	90
Galveston, Tex.	60.1	80	12.1	5	10	10	75	71.9	73	9.9	5	10	10	75
Jacksonville, Fla.	60.5	71	12.9	0	0	5	95	78.0	73	10.7	0	0	0	100
Los Angeles, Calif.	57.4	52	5.5	10	10	20	60	67.4	63	5.0	2	1	10	87
Minneapolis, Minn.	26.1	62	12.1	5	42	40	13	63.0	53	10.7	1	5	21	73
Minot, N. Dak.	19.0	77	9.6	45	30	20	5	56.0	61	8.6	2	3	35	60
Montreal, Que.	24.3	..	15.4	35	55	8	2	59.4	..	11.8	10	12	33	45
New Orleans, La.	60.1	70	7.3	5	20	35	40	78.5	65	6.1	1	6	20	73
New York, N. Y.	38.1	63	20.5	5	60	30	5	66.6	65	14.9	1	5	19	75
Newark, N. J.	38.6	66	19.7	20	25	30	25	66.0	70	15.4	5	10	10	75
Philadelphia, Pa.	40.2	59	10.8	10	50	25	15	68.5	63	8.6	5	10	25	60
Pittsburgh, Pa.	38.5	67	12.6	20	55	17	8	67.1	67	9.0	0	2	13	85
Portland, Ore.	44.7	76	6.3	13	39	32	16	61.4	63	5.8	2	7	34	57
San Francisco, Calif	53.2	70	8.0	10	22	38	30	59.0	72	9.9	2	12	18	68
Seattle, Wash.	43.7	75	9.1	5	32	31	32	58.2	69	7.8	0	7	27	66
St. Louis, Mo.	40.8	65	13.9	14	20	33	33	71.0	65	11.2	0	0	20	80
Tampa, Fla.	64.6	70	6.7	0	2	8	90	79.5	72	6.1	0	0	0	100
Vancouver, B. C.	40.5	..	4.7	27	18	37	18	56.9	..	4.4	2	2	50	46
Waco, Tex.	55.0	65	8.7	5	5	5	85	78.3	61	8.0	0	0	5	95
Winnipeg, Manit.	13.2	..	12.9	44	45	9	2	56.5	..	12.9	10	24	26	40

general, milder than ours. Hill and Campbell (8, pp. 111 and 117) make these observations: "In this country (referring to England) the average person is generally dressed unsuitably for the heat of summer, when sometimes the conditions are tropical. . . . It is not surprising that deaths from heat stroke occur, seeing that in many cases winter clothing is worn. While in tropical climates civilized man often removes his coat in public on excessively warm days, such an event is very rare in London on a similarly oppressive day. Habit and fashion rule, and not the subject's feelings."

According to Fig. 5, the representative types of underclothing worn in those parts of the United States where the climate is comparable to that of England are medium weight cotton in winter and light weight cotton in summer.

In making our comparisons, we must not forget that both outdoor and indoor atmospheres in England are generally more humid than ours, on account of the

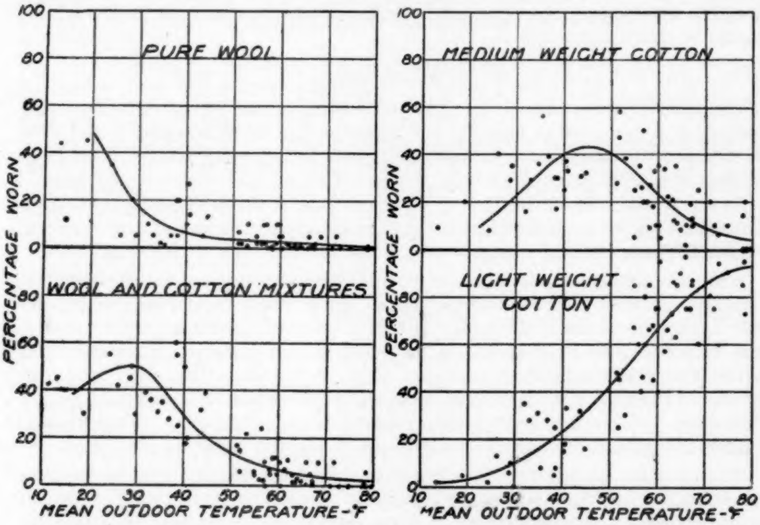

FIG. 5. UNDERCLOTHING IN RELATION TO SEASONAL OUTDOOR TEMPERATURE

moist climate and the fact that the air entering the buildings does not have to be heated through such a wide range as it does in our colder climates. Under these conditions, woolen underwear is preferable to cotton.

Adaptation to climate is also an important factor. The intense summer heat to which people in the greater part of the United States are accustomed causes them to require a higher temperature in winter; whereas the intense cold of winter is largely mitigated by artificial heating indoors. Moreover, adaptation to high temperatures may continue during the heating season, on account of the overheating of rooms in mild weather. This condition results, not so much from choice, as from the circumstance that, owing to the changeableness of the American climate, the heating systems have to be designed with ample capacity for making buildings reasonably comfortable during the coldest weather. With the ordinary house heating systems, especially with those using steam as the

heating medium, it is difficult to prevent overheating in autumn and spring and in mild winter weather.

In England, adaptation during the winter season works in just the opposite manner. The heating facilities there are not, as a general rule, so adequate as ours, because there is less need for elaborate systems in their climate. Consequently the English people become accustomed to cooler room temperatures, which they partly offset by appropriate clothing.

The difference in heating methods employed in the two countries introduces another factor which merits consideration. With the representative English radiant heating systems, such as coal and gas fires, rooms feel comfortable at a lower air temperature than they do when heated by the American convection methods—steam or hot water radiators and warm air systems. Vernon gave evidence of this (9, p. 56) when he found that rooms heated by coal and gas fires felt comfortably warm at a temperature 7 deg cooler than similar rooms heated by convection methods.

SUMMARY

The summer comfort zone for men and women in the United States wearing customary warm weather clothing has been found to lie between 64 F and 79 F on the effective temperature scale. The probable optimum is 70.5 F effective temperature. On the average, these values are about 4.5 F effective temperature higher than those found in winter, when customary winter clothing was worn. The difference is ascribed to adaptation to seasonal weather as well as to seasonal variation in the clothing worn.

A definite relationship is shown to exist between the seasonal outdoor temperature and the kind of underclothing worn. Pure woolen underwear is popular only in those localities of the United States in which the average winter temperature falls below 20 F; wool and cotton mixtures are preferred in cities having winter temperatures between 20 F and 37 F; in warmer cities, where the temperature ranges from 38 F to 50 F, medium weight cotton is worn by the majority; when the average temperature exceeds 50 F, whether in summer or in winter, the greater part of the underclothing worn is light weight cotton.

The optimum temperature was found to follow the average monthly outdoor temperature more closely than the prevailing outdoor temperature. It remained at approximately the same value in July, August and September, probably because the average monthly temperature did not vary much, although the prevailing outdoor temperature ranged from 70 F to 99.5 F. A decrease in the optimum temperature became apparent only when the prevailing outdoor temperature fell to 66 F, which is below the customary room temperature in the United States for summer and winter.

Crowding the experimental chamber lowered the optimum effective temperature from 70.8 F when the gross floor area per occupant was 44 sq ft and the air space 380 cu ft to 69.4 F when the floor area was reduced to 14 sq ft and the air space to 120 cu ft per occupant.

For practical use, the summer and winter comfort zones were superimposed upon the effective temperature chart designed at the Pittsburgh Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. By means of this new chart, the relative comfort of ordinary indoor conditions can be determined from the dry- and wet-bulb temperatures of the air.

An attempt is made to explain the difference between the English and American comfort standards by differences in climate, clothing and heating methods.

We wish to express our thanks to the clothing firms which were kind enough to furnish us with their data.

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DISCUSSION

T. J. Duffield (Written): The establishment of the summer comfort zone for adults normally clothed and slightly active in still air by Mr. Yaglou and Professor Drinker fills one of the large gaps heretofore existing in our knowledge of comfort reactions of human subjects under different effective temperatures. The Society is indebted to them for thus complementing the study earlier made at the Society's Research Laboratory.

I am not going to take the time to discuss all the meritorious features of this research. The report speaks for itself. One phase of the study has, however, not received in the past the consideration it has deserved and still offers opportunity for more extensive study. I refer to the influence of occupancy on optimum temperature. This is a matter which involves the proportion of body heat lost by radiation and is probably of prime importance in providing comfortable air conditions in theatres, and possibly also in school rooms. The effect of the proximity of one's neighbors in preventing normal body heat loss by radiation may be an important factor in the maintenance of health and comfort among school children.

In the classroom, the radiant heat loss is further affected by the fact that one or more walls is an outside wall of the building where infiltration and structural heat losses create surface temperatures quite different from that of an inside wall. The problem is further complicated by the fact that two-thirds of one of the outside walls is of glass, which on the one hand does not offer the same resistance to the transmission of radiant heat as does a wall, and

on the other assumes surface temperatures which vary with the indoor and outdoor air temperatures, the wind velocity and the rate of air movement indoors, as has been shown in another paper to be presented at this meeting.[7]

In this connection, I should like to call attention to a report that has recently been published by the Smithsonian Institution on the subject of body heat loss by radiation.[8] This study, which was made under the direction of the Secretary of the Institution, one of the world's authorities on radiation and its measurement, at the suggestion of and with some financial assistance from the *New York Commission on Ventilation*, was of a preliminary nature and did not cover a sufficiently wide range of temperatures and humidities to establish the ratio of radiation to total body heat loss under various conditions. At the normal indoor temperature of 70 F in still air, with the subjects normally clothed and at rest, it was found that the proportionate heat loss by the various methods was:

1. Radiation	46 per cent
2. Convection	30 per cent
3. Evaporation	24 per cent

Increasing the air motion reduced the percentage loss by radiation both with human subjects and with a calorimeter on which certain clock observations were made. However, the importance of radiation is obvious and the desirability of its further study clearly indicated.

There is some question in my mind as to the validity of the comparison of the winter comfort zone established in Pittsburgh in 1922 and the summer zone determined in Boston in 1927. I do not recall whether male or female subjects predominated in the Pittsburgh Study nor whether the proportionate representation of the sexes was the same as in the Boston study. Although I was out of the country for several years before and after the Pittsburgh study, I think I have observed, on the part of the adult female, since my return in 1925, a decided tendency to wear less and less, and that more and more dependence has been placed on the product of the silk worm (or man's imitation thereof) than on that of the sheep.

Clothing habits among women have changed in the period between the two studies and as far as indoor clothing is concerned, have tended toward uniformity throughout the year. Men, on the other hand, have not altered their customs to any degree between the two studies and many still take the seasons into consideration in deciding what clothes to wear. This is illustrated by the fact that in the Boston study some of the male subjects wore Palm Beach suits. All this leads to the suggestion that I have previously passed on to Mr. Houghten, namely, that summer and winter comfort zones for men and women be determined independently. This would greatly help, I am certain, to clear up some of the difficulties we experience in our school ventilation studies. It goes without saying that a study of the comfort reactions of children of school age and a determination of their comfort zones is necessary to complete the picture.

In our studies of schoolroom ventilation we are confronted with the spectacle of a room in which the air conditions are dictated insofar as possible by the teacher, who, by reason of differences of age, metabolism and clothing habits,

[7] Frost and Condensation on Windows, by L. W. Leonhard and J. A. Grant, p. 295.
[8] A Study of Body Radiation, L. B. Aldrich, *Smithsonian Miscellaneous Collections*, Vol. 81, No. 6, Dec. 1928.

is comfortable at quite different effective temperatures than are the 40 or more pupils who share the room. It would be quite logical, I think, to create in the classroom conditions found most comfortable for the pupils and then let the teacher dress accordingly.

In closing, I want to congratulate the authors for having collected information on clothing habits to explain differences in comfort reactions at different seasons and in different latitudes. It should be pointed out, I feel, that all the data here presented for the different weights of underwear are of the masculine gender.

E. S. HALLETT (WRITTEN): This paper by Yaglou and Drinker is very timely. We have come to the point where we must do something more than learn the reaction of test individuals in a laboratory, we must meet the situation with requisite equipment to preserve living and working temperature conditions during the hot weather in most of our American cities. We have a means of knowing what the average man and woman thinks about it as indicated by the crowds that pack the great theatres two or three times every day during the hot weather, mostly to escape the heat everywhere else.

These experiments disclose the fact that persons differ quite widely in their sense of comfort as given in Table 1, while the largest number are very comfortable at 70 F to 71 F. There are many others who vote for temperatures from 66 F to 76 F. It is these edge bands of the wave that give us the grief in operating schools at the fixed temperature of 70 F. The authors do not mention the situation of teachers and others who live in overheated apartments until a high temperature *habit* has been formed that is far more difficult to break than the cigarette habit. They do not suffer from the heat.

In these tests all other conditions but temperature, humidity and air motion have been eliminated as is necessary in a scientific test, yet in practice we find discomfort due to other things charged up against the temperature. A striking case recently came up in St. Louis in a one story school with a concrete beamed ceiling. Complaint was made of the sound reverberation, and it was mostly too cold at 70 F as well. Sound proof felt was installed on the beams and immediately all expressed surprise at the sense of warmth and comfort that was experienced. The thermometer readings did not change. It may appear that the comfort zone is composed of three elements everywhere except in a schoolroom. This test did not attempt that problem.

The facts brought out in Table 3 are illuminating and valuable. The authors think the clothing is responsible for most of the difference. I wonder if the customs of the country and the mental state may not have something to do with the seasonal difference of comfort.

Attention ought to be directed to the suffering and loss of efficiency occasioned by this country. The St. Louis schools close by rule when the outside temperature reaches 90 F and it is apparent that useful work stops at about 80 F. It is easily within the facts to say that 5 per cent of the school activities of the school system are lost each year by high temperatures. The ultimate purpose of this paper must point a solution to the problem. It is an appeal to this Society.

W. H. CARRIER: One of the very interesting points in this paper which Mr. Yaglou gave us scarcely sufficient time on, in my opinion, in view of its importance, was the emphasis of the location of this summer comfort line, and its relation to the winter comfort line. I do not know how much of this is due to what we want to call acclimatization, and how much of it is due to the lighter

clothing, and due to the fact that people entering air-conditioned auditoriums, etc., from a hot outdoor condition, are covered more or less with perspiration, the clothing is damp, and where there is an initial condition that is apt to be a little severe if the temperature is at all low.

All of this requires, I think, as Mr. Yaglou pointed out, a higher plane of summer temperature conditions in places of public assembly than perhaps the optimum condition which would be required for people working in an industrial establishment or office building for 6 or 8 hours continuously.

The results are very interesting, and I believe very useful to the ventilating profession, especially those of us who are interested in securing a reasonable and desirable temperature in places of public assembly.

C. P. YAGLOU: Concerning Mr. Carrier's point on acclimatization, I should like to make clear that the most probable optimum effective temperature for summer, as fixed by our experiments, applies chiefly to cases in which the human body has reached thermal equilibrium with the surrounding air. The case is quite different when one enters a cool room on a hot summer day, though the room may be at the optimum temperature. Immediately upon entering, one is liable to experience an intense chill, or the shock which Mr. Lyle discussed this morning. However, after about 2 hours exposure, this optimum temperature condition will probably be found quite satisfactory by the average person.

In some of our experiments the subjects did not enter the psychrometric room immediately from the hot outdoor atmosphere, but they first spent about 20 min in another room, the temperature of which was between the prevailing outdoor temperature and the optimum temperature, at which the psychrometric room was maintained. Under these conditions they did not report sensations of chill when they entered the psychrometric room, although they said that the room felt cool.

The object of cooling theaters in summer is not to reduce the temperature to the optimum degree, but it is to maintain therein a reasonably comfortable temperature, and at the same time avoid sensations of chill or of intense heat in entering and leaving the building. This may, possibly, be accomplished by keeping the auditorium at a temperature half-way between the prevailing outdoor temperature and the optimum temperature for continuous exposure. It may also prove advantageous to keep the theater entrances and lobbies at a temperature half-way between that prevailing out of doors and that in the auditorium. All these indoor conditions lie within the summer comfort zone.

In regard to Mr. Duffield's questions, the factors of sex, age, occupation, clothing, etc., they all exert an influence upon one's sensations of comfort. For these reasons we have determined a comfort zone, not a comfort line, and we chose a group of subjects which was about the same as in the Pittsburgh experiments. Our range of air movement was also the same, as in the Pittsburgh experiments, namely 15 to 25 fpm, according to the Kata-thermometer.

As to the possibility of women wearing lighter clothing in 1926 and 1927 than in 1923, when the Pittsburgh experiments were carried out, I do not believe that this could be of much consequence, since, in recent years, women have really been wearing little clothing and they probably cannot get away with anything less.

No. 832

LOW HUMIDITY PSYCHROMETRIC CHARTS

By Malcolm C. W. Tomlinson, Westtown, Pa.

MEMBER

FOUR CHARTS are presented which can be read to the tenth of a per cent relative humidity over a dry-bulb range of —30 to 170 F, and a relative humidity range of 0 to 15 per cent. New developments in the electrical industry, by which better electrical characteristics are secured through air conditioning at very low humidities, and the possibility of using the same humidities, in any locality, to cure diseases now treated by sending patients to dry climates justify the effort.

Dehumidification at low relative humidities, in the neighborhood of 10 per cent, have been used in the electrical industry for over three years. The success of this venture led investigators, during the past year, into the field of humidities lower than 1 per cent. It is now apparent not only that hygroscopic insulation for sensitive electrical equipment can be protected best, in the process of manufacture, against moisture regain by this latest form of dehumidification, but also that much better insulation resistance, conductance and even capacity can thus be secured. Experience has also shown that workmen, subjected to these very low relative humidities, are quite free from colds. Since medical specialists have found it beneficial to send patients with certain types of diseases, especially pulmonary, to live in dry climates it is not unreasonable to predict that, in any climate, it will not be long before hospitals and sanitariums will be prepared to furnish suitable atmospheric conditions to meet the needs of a wide variety of diseases. This service should especially appeal to those who find it necessary to remain close to their business and social ties.

At present only one psychrometric chart, the Bulkeley,[1] can be read to the fraction of a per cent and, at that, only over a comparative short range on dry-bulb temperature. The four charts, Figs. 1, 2, 3 and 4, presented herewith, now make it possible to read the fraction of any per cent relative humidity below 15 per cent and between —30 and 170 deg dry-bulb temperature on the Fahrenheit scale. They also supply much of the additional data usually found on such charts. Although the maximum relative humidity shown is quite low this disadvantage is small as those who are working at very low relative humidities will seldom need to go above the range covered.

[1] Trans. A.S.H.&V.E., Vol. 32, 1926.

Presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Chicago, January, 1929.

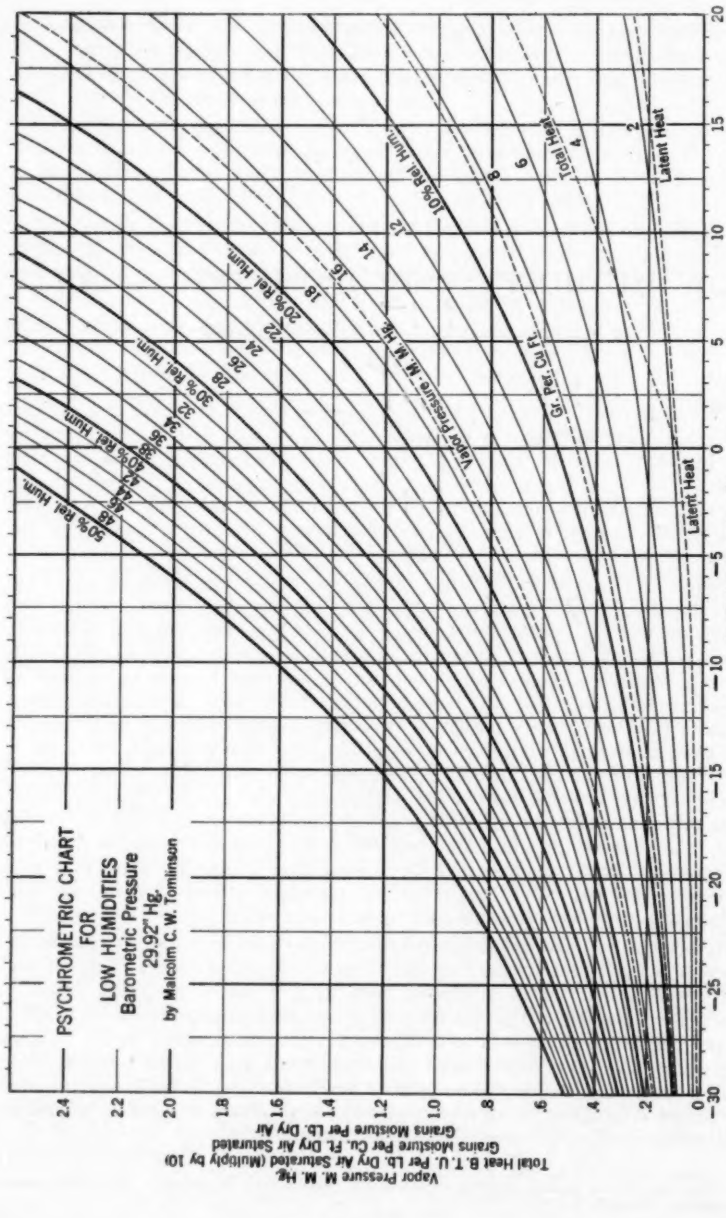

FIG. 1. TOMLINSON PSYCHROMETRIC CHART FOR LOW HUMIDITIES

Many forms of psychrometric charts are available. One of the earliest was prepared by Prof. H. L. Parr.[2] Others in more or less general use include the Willis H. Carrier[3] charts for ordinary and high dry-bulb temperatures; the charts of Louis A. Harding,[4] of William H. Grosvenor,[5] of H. D. Tiemann[6] and the rather unique chart by Claude A. Bulkeley to which reference has already been made. This latter chart, copyrighted by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, is the only psychrometric chart which uses a form which is practically semilogarithmic. Certain advantages are thus obtained at the expense, naturally, of others. It might be interesting, in this connection to point out the main advantage of semilogarithmic charts. They furnish the best comparison of trends, in graphing, which can be secured where the various sets of statistics presented have numerical values of different magnitudes. As an example suppose the number of radiators produced each week, by a given manufacturer, were plotted through a period of a year and then, on the same chart, plot the sales in dollars. Graph forms laid out on the rectangular coordinate scale would not give the true trend, in this case, due to the fact that the average radiator costs much more than one dollar. The same information plotted on the semilogarithmic chart would make possible a true picture of how close production and sales were in step one with the other. It therefore seems obvious that the Bulkeley chart has a field of usefulness, in plotting trends, not generally realized. At the same time it is not as satisfactory, equivalent sizes of charts being considered, as some of the other psychrometric charts where one desires to work above 10 per cent relative humidity or below 50 F dry bulb. This fact, as well as the general use of the Carrier Charts on the part of engineers, has led the author to adopt the Carrier form.

The basic data used are, of course, vital to the accuracy of such charts. The water vapor pressures used were taken from the International Critical Tables.[7] The moisture in dry air, which has been saturated, as well as the volume at any given temperature per pound of dry air saturated were calculated by means of the thermodynamic formula:

$$pv = BT \quad \text{where } \begin{aligned} p &= \text{pressure in square feet} \\ v &= \text{volume in cubic feet} \\ T &= \text{absolute temperature} \\ B &= \text{gas constant} = 53.34 \text{ for air} \\ &= 85.7 \text{ for water vapor at low pressures.} \end{aligned}$$

The volume of dry air and the total heat data were obtained in Goodenough's Tables.[8] The heat of the liquid is not included in the values charted. This is in keeping with the usual psychrometric chart practice. The wet-bulb temperatures were calculated by means of the latent and total heat data given in the Goodenough Tables. The wet-bulb lines plotted from this data will be found,

[2] Engineering Thermodynamics, C. E. Lucke, 1912.
[3] *Trans. A.S.M.E.*, Vol. 33, 1911.
[4] *Mechanical Equipment of Buildings*, Vols. 1 & 2, L. A. Harding and A. C. Willard, 1917.
[5] *Trans. Am. Inst. Chem. Eng.*, Vol. 1, 1908.
[6] Forest Service Bulletin No. 104, 1912.
[7] International Critical Tables—Vol. 3.
[8] Table 6. Mixtures of Air and Saturated Water Vapor, Properties of Steam and Ammonia, G. A. Goodenough, 1915

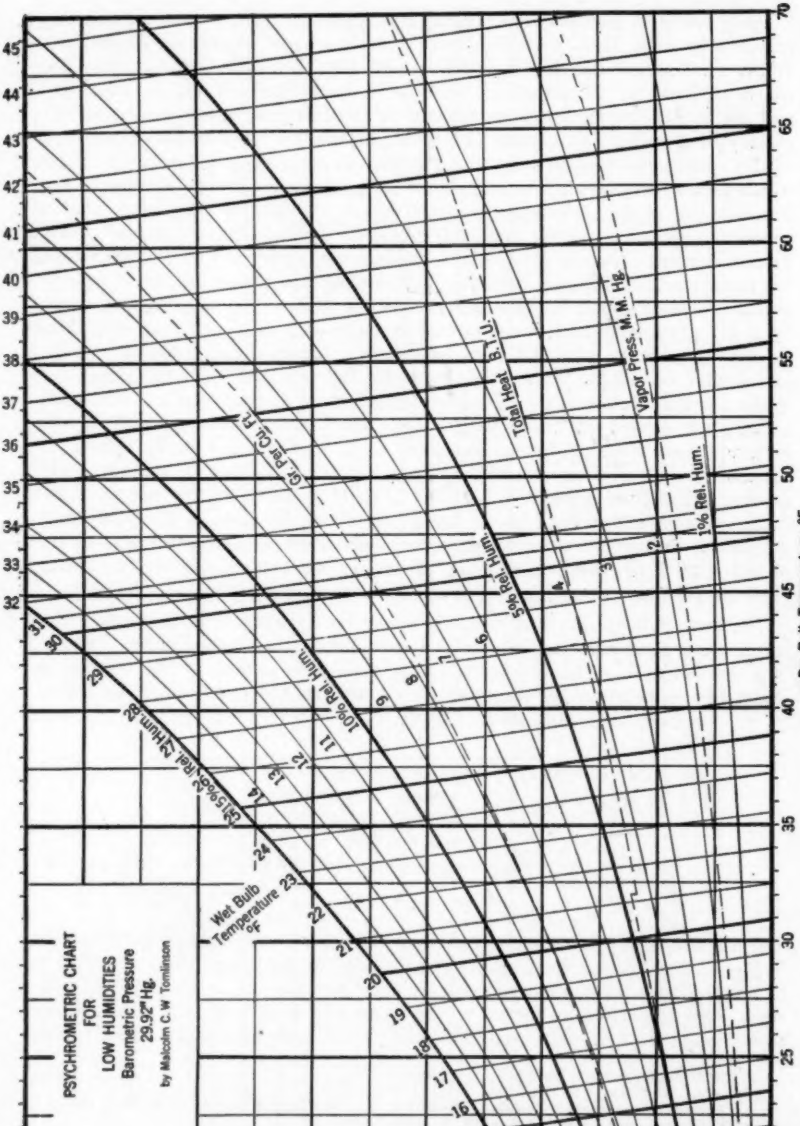

Fig. 2. Tomlinson Psychrometric Chart for Low Humidities

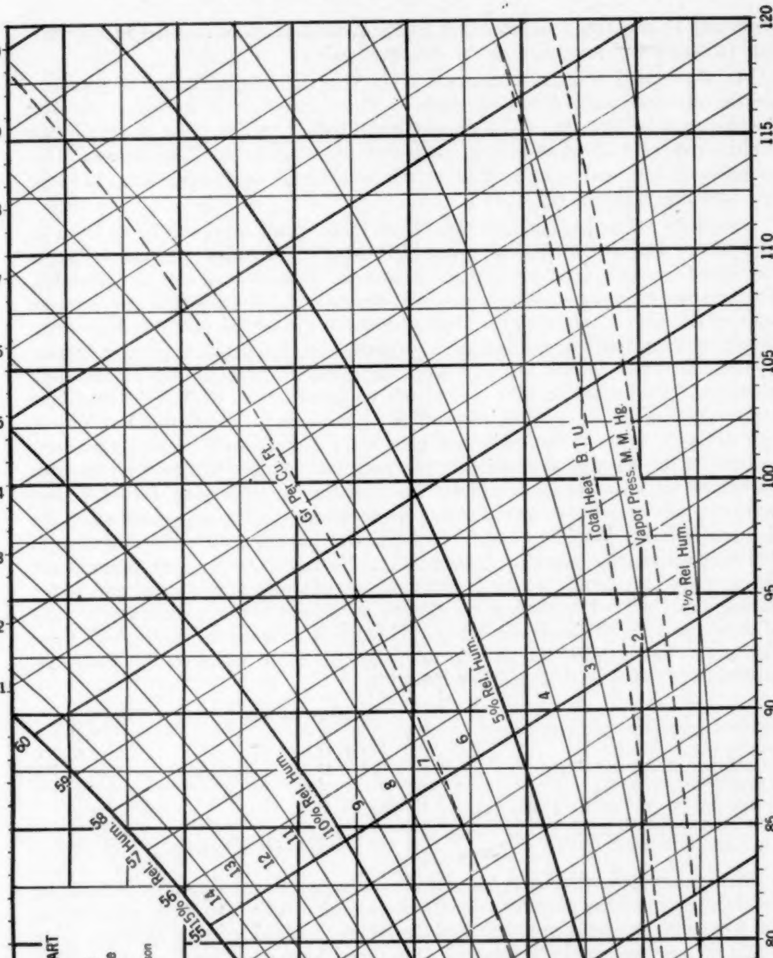

FIG. 3. TOMLINSON PSYCHROMETRIC CHART FOR LOW HUMIDITIES

especially in the Fig. 2, much closer to the theoretical than is usual in psychrometric charts due, naturally, to the enlarged scale.

For the benefit of those who are not familiar with the technique of psychrometric observations it seems advisable to direct attention to the paper on The Temperature of Evaporation by Carrier and Lindsay.[9] The effect of velocity on the error in wet-bulb observations is there presented in an excellent manner. It is to be hoped that further investigations will be made along identical lines to cover wet-bulb temperatures below 44 F.

Simplicity has been sought in this effort. Therefore curves showing the volume of dry and of saturated air were eliminated. The latter data can easily be calculated from the moisture per pound and per cubic foot of dry air saturated. Furthermore conversion factors may be obtained, easily, by which any one can convert the units charted into equivalent units in the metric or English systems. While the two charts do not furnish a saturated (or 100 per cent relative humidity) curve, much of the essential data are given for dry air which has been saturated. For example, take a dry-bulb temperature of 45 F and a wet-bulb temperature of 30 F. At the intersection of these lines the relative humidity is read from the nearest relative humidity curves. Directly to the left of this intersection, at the margin, the moisture per pound of dry air will be read for this particular air-water vapor mixture. The moisture at saturation, for 45 F dry-bulb, can now be calculated by dividing the moisture reading just obtained by the per cent relative humidity found on the chart. The result is the moisture at 100 per cent relative humidity (saturation). For the same dry-bulb temperature the saturated air conditions as to moisture per cubic foot, total heat and vapor pressure can be read on the margin directly to the left of the intersection of each of these curves with the dry-bulb line. The degree of accuracy with which these charts can be read may be seen in the following table based on the foregoing problem and using a 13 in. x 20 in. chart:

	Observed Reading		Calculated
Relative humidity—per cent	8.25		8.45
Grains per lb dry air	3.58		3.43
Grains per lb dry air (sat)	43.39	(calculated)	44.11
Vapor press—mm Hg	7.60		7.62
Grains per cu ft dry air (sat)	3.43		3.43
Total heat—Btu	17.60		17.59
Cu ft per lb dry air (sat)	12.65	(calculated)	12.85

Early psychrometric charts antedated satisfactory instruments for measuring the relative humidity directly. Instruments which were available operated on the wet-bulb, dry-bulb principle and the reader needed a table or chart from which the readings could be interpreted in terms of the relative humidity. Most of these instruments were inaccurate.

The increasing use of the A. S. H. V. E. comfort data and the use of humidification for industrial processes, where relative humidities are measured much more accurately, have created a demand for charts and instruments of a more reliable type. The charts presented in this paper meet this situation for the low humidity field.

Accurate and reliable indicators, recorders and controllers, which measure relative humidities directly, are now available. Similar instruments for precise

[9] *Trans. A.S.M.E.*, Vol. 46, 1924.

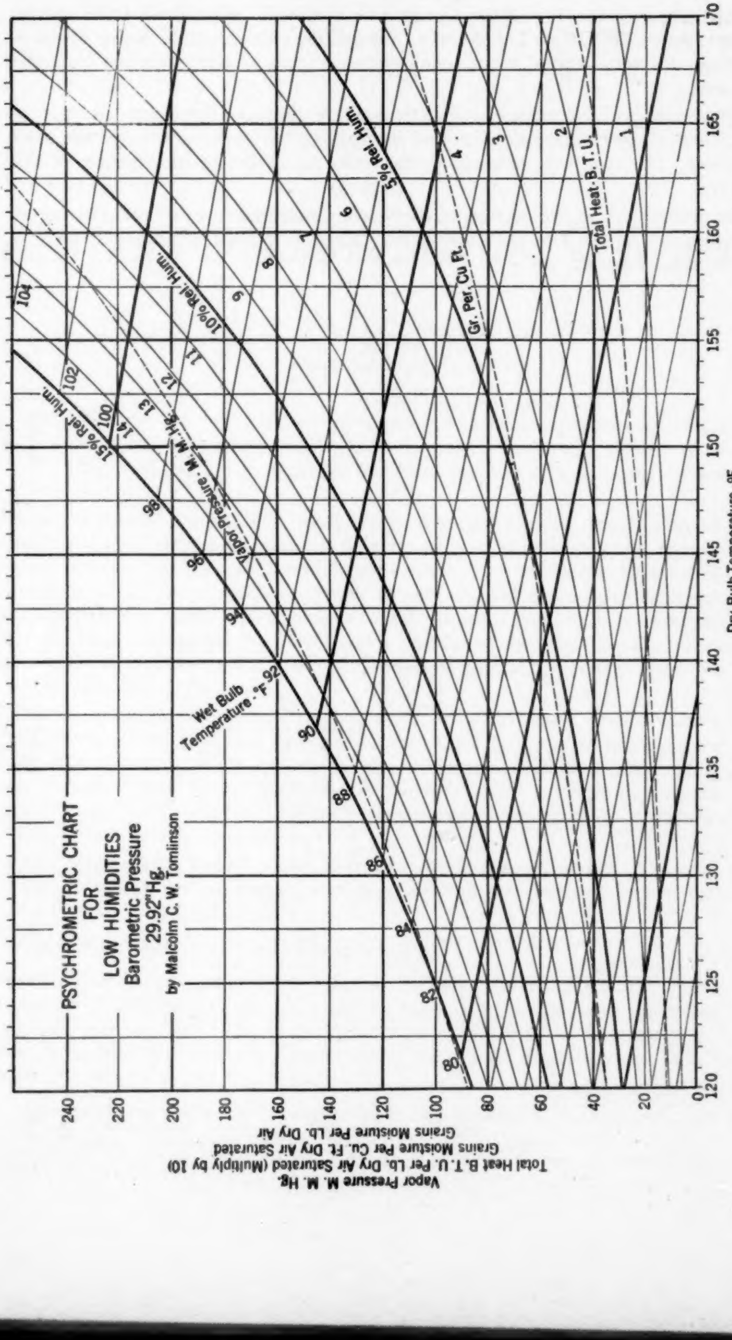

FIG. 4. TOMLINSON PSYCHROMETRIC CHART FOR LOW HUMIDITIES

measurement of low humidities can also be obtained. With these latter instruments, the charts (Figs. 1, 2, 3 and 4) furnish the only available means of determining the wet- and dry-bulb temperatures from charted relative humidity records.

Previous texts and papers have so thoroughly discussed the theory and the application of psychrometric charts that it would be burdensome to cover the same ground. The reader is referred to the notes herewith for information of this nature.

In closing the author wishes to express his indebtedness to Prof. George A. Goodenough whose fine air-water-vapor tables have been a great help in preparing these charts.

No. 833

FROST AND CONDENSATION ON WINDOWS

By L. W. LEONHARD[1] AND J. A. GRANT,[1] DETROIT, MICH.

NON-MEMBERS

MOISTURE on windows either in the form of condensation or frost causes no great annoyance in the majority of instances. However, in some cases, this formation on the windows is not only annoying but very detrimental.

There are several reasons why the problem has grown more acute. The advantages of higher humidity of the air have resulted in the development and use of means for maintaining a higher moisture content than formerly. Better construction, at the window openings, has materially reduced the rate of exchange of air between the inside and outside of buildings, thus causing a building up of the moisture content of the inside air. Buildings nowadays have a much greater window area, and the amount of condensation is dependent upon the amount of window surface.

All of these factors have contributed to render the conditions more favorable for the deposition of moisture upon cold window surfaces. Not only has the amount or quantity of water thus deposited on the coldest days of the heating season been increased, but the formation of moisture occurs on more days of the season, because the favorable conditions enumerated above result in condensation at a higher outside temperature.

In this paper, these factors which influence the formation of frost and condensation on windows are discussed and some methods for lessening or preventing this formation are suggested.

The Inside Temperature of the Window Surfaces

In order to determine quantitative values for some of the principal factors that influence the inside temperatures of windows, some experiments were conducted during the winter of 1926-1927, in the Mechanical Engineering Laboratory of the University of Michigan with the cooperation of Prof. J. E. Emswiler[2] and W. C. Randall.[3]

[1] Research Engineer, Detroit Steel Products Co., Detroit, Michigan.
[2] Professor of Mechanical Engineering, University of Michigan.
[3] Chief Engineer, Detroit Steel Products Company, Detroit, Michigan.

Presented at the Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, January, 1929.

A small window two panes wide and three high was placed in one end of a cold box of the laboratory refrigerating plant. Any desired low temperature could be maintained on the one side of the window with an ordinary room temperature on the other. Small thermocouples, flattened out very thin, were glued to the glass or other surface to be examined, whereby the temperature of those surfaces could be ascertained. A variable speed fan was located in the box, so that when desired, the cold air could be blown over the surfaces of the window, thus simulating wind conditions. The humidity of the air on the warm side could be varied by releasing steam into the room.

The objective of the tests was to determine the ratio of the difference between the temperatures of the inside surface of the glass, or other surface examined, and that of the cold air, to the difference between the temperatures of the warm and cold air. This ratio or factor may be expressed mathematically as follows:

$$f = \frac{\text{inside surface temperature} - \text{cold air temperature}}{\text{warm air temperature} - \text{cold air temperature}}$$

Thus, if the temperature of the warm air is 70 F, the temperature of the cold air 10 F, and that of the inside surface 30 F, then

$$f = \frac{30 - 10}{70 - 10} = \frac{20}{60} = 0.33$$

The values of this factor, or ratio, defined previously and determined by the experiments are about as shown in Table 1.

TABLE 1

Item	Material	Conditions	Value of f
1.	Single Glass—¼-in. Plate	No Wind	0.50
2.	Single Glass—¼-in. Plate	With Wind	0.25
3.	⅛-in. Double Glass with ⅛-in. Air Space	With Wind	0.48
4.	⅛-in. Double Glass in Contact	With Wind	0.35

In the table, values for the factor f are given for the conditions of *No Wind* and *With Wind*. *No Wind* means that the fan on the cold side of the window was not in operation.

As is to be expected, winds or air velocities of different magnitudes will produce different temperatures on the inside surfaces of the window. However, it was found that whereas a small wind velocity caused a rapid reduction of the inside surface temperatures, the rate of decrease did not keep pace with increase of wind velocity, and at about 8 mph or more, there was but little further effect produced. The relations between wind velocity and the ratio, or factor f, are shown in Fig. 1.

For no wind, the temperature of the inside surface of the glass is just about half way between the temperature of the outside cold air and that of the inside warm air. For a moderately high wind velocity, the inside glass temperature is only about one-quarter way between the two air temperatures. Thus, if the cold air is at 10 F and the warm air at 70 F, the inside temperature of the glass for no wind is $10 + \frac{1}{2}(70 - 10) = 40$ F, and with a wind of 8 or 10 mph, the inside temperature of the glass is only $10 + \frac{1}{4}(70 - 10) = 25$ F. It is, therefore, seen that the wind is a very strong contributing factor in the formation of frost and condensation. This statement is borne out by the observed fact

that frost and condensation appear first and are heaviest on those windows that are located on the windward or exposed side of a structure.

The curves of Figs. 2 and 3 show temperature relations between inside and outside air, dew-point of inside air, and the inside surface temperature of the glass for numerous temperature conditions with and without wind action upon the outside surface. It should be noted that the horizontal scale does not represent equal time intervals, but merely indicates test conditions. These curves have been plotted from actual test data and show how closely the temperatures registered by the thermocouples agreed with the observed condition of the glass surface; for example, when the inside glass temperature became equal to or lower than the dew-point temperature of the inside air, as registered by the thermocouple reading, moisture was observed to be deposited upon the surface. The

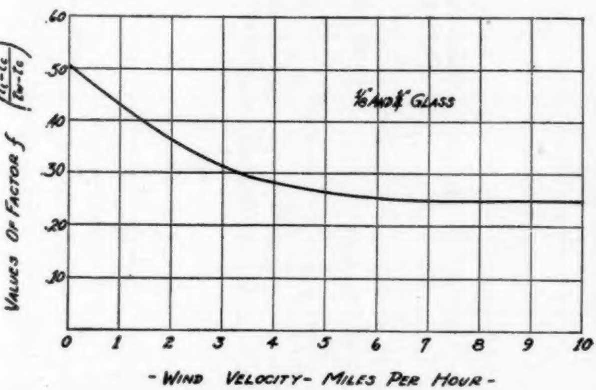

FIG. 1. RELATION OF FACTOR "f" *to* WIND VELOCITY

same was true for other observed conditions indicated by the vertical lines intersecting the curves. Fig. 2 also shows how the inside surface temperature of the glass for the *No Wind* condition is practically midway between the inside and outside air temperatures but for the condition of wind action upon the outside surface, Fig. 3, shows the inside glass temperature to be much closer to that of the outside air, approximately three-fourths of the distance from the inside air temperature curve to that of the outside air.

The Formation of Condensation on Windows

The factors that chiefly influence the deposition of moisture and the formation of frost are two in number:

1. The dew-point temperature of the inside air.
2. The temperature of the inside or room surface of the glass.

The dew-point temperature is determined solely by the moisture content, or more properly speaking, the steam content, of the air. The moisture content of the air is usually expressed in terms of the relative humidity in conjunction with the temperature of the air. Thus, if the temperature and the relative humid-

ity are definitely known, the actual weight of water vapor, or steam, contained in 1 cu ft space, as well as the dew-point temperature, can be determined.

The control of the moisture content of the air offers the best possible means of controlling the dew-point temperature and, consequently, the deposition of frost and condensation. If the air is cooled, without the addition or subtraction of any moisture, the dew-point temperature remains the same as before. If the moisture content is increased, the dew-point temperature will be raised, practically, regardless of what the room temperature may be, and, again, if the

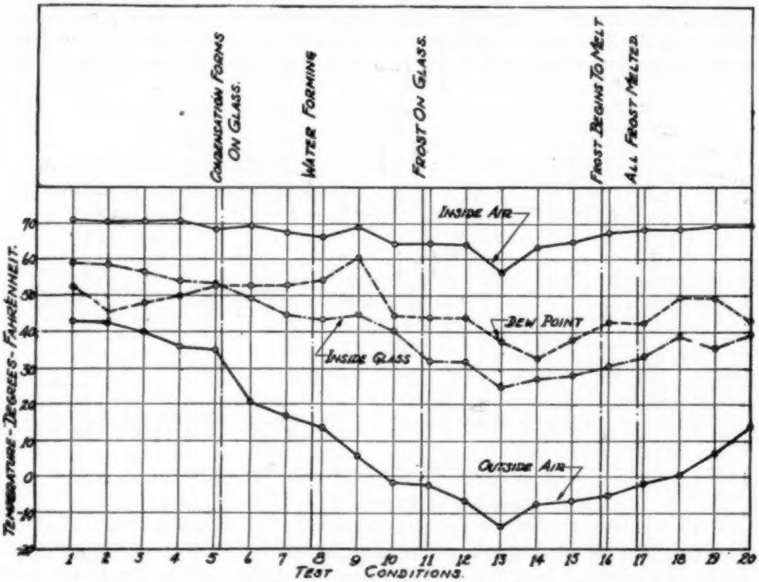

FIG. 2. TEMPERATURE RELATIONS FOR CONDITION OF "NO WIND"

moisture content of the air be reduced, the dew-point temperature will likewise be lowered.

If it were always possible to keep the temperature of the inside glass surface above the dew-point temperature of the inside air, frost or condensation would never form upon the glass. The inside temperature of the glass is dependent upon a number of factors. It is evident that it will be influenced by both the inside and outside temperature of the air, and that it will have a value somewhere between these two. There will be a certain temperature gradient, or rate of temperature drop, accompanying the flow of heat from the warm air inside to the cold air outside. The character of the temperature drop will be determined by the nature of the resistances offered to the heat flow. The resistances to heat flow consist of a film of air in intimate contact with the glass on the warm side; the material of the glass itself; and a film of air in intimate contact with the glass on the outside. It is a very easy matter to control the inside air temperature, but

increasing the air temperature does not affect a corresponding increase in the inside surface temperature of the glass, and for this reason, does not offer a practical method of eliminating the formation of frost and condensation.

Constructional Means of Preventing Frost and Condensation

The most effective means of increasing the inside glass temperature is to increase the resistance to the flow of heat through the glass itself, such as might

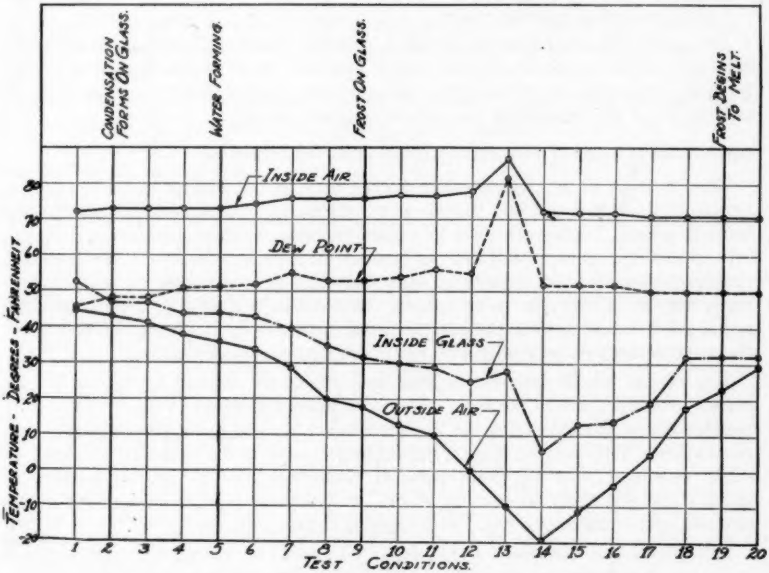

FIG. 3. TEMPERATURE RELATIONS FOR CONDITION OF "WIND"

result from the use of double glazing, double windows, or by applying some transparent insulating material in the path of the heat flow.

The prevention of frost and condensation by constructional means is more applicable where high humidities must necessarily prevail. It is to be noted, in this connection, that the benefit to be derived from even a few degrees increase of the inside glass temperature is very great, considering the fact that condensation and frost appear only on the coldest days of winter.

Double glazing, with an air space between, shows quite an advantage over single glazing as far as restraining the formation of frost and condensation is concerned, but there are also some important disadvantages which should be kept in mind:

1. The air contained between the two panes of glass must be practically dry air and a permanent air-tight seal around the edges between the two panes is necessary to keep moisture or moist air from penetrating into this confined space; otherwise condensation or frost will be found to accumulate

upon the inside surface of the outside pane even for moderate temperature conditions which would not cause frost or condensation to form upon the inside surface of single glazed windows.

2. It is practically impossible, by ordinary glazing methods employing putty, etc., to make a permanent air-tight chamber between the two panes of glass. Temperature changes due to changing weather conditions cause an expansion or contraction of the air space between the panes of glass. This results in a breathing which draws moist air into the confined space and condensation takes place.

The same disadvantages hold true for double windows and storm windows. However, such installations seem more practical than double glazed windows because provisions can be made for cleaning the glass surfaces between the two windows and for ventilating the intervening air space.

Operational Means of Preventing Frost and Condensation

Condensation to an objectionable degree may be considered as an emergency situation, since it is not one that must be contended with continually during the heating season, but occurs only in rather extreme weather conditions. Therefore, if this emergency condition can be dealt with by a reduction of the moisture content of the air, the benefits of high humidity can be dispensed with temporarily for the advantages to be gained. If some dehumidifying device could be produced, it would answer the purpose in those emergency situations. However, the same result can be attained by more simple and obvious means.

The rate at which moisture is generated should be reduced by cutting off all moisture sources where possible. Then the moisture content can be reduced by natural means as exchanging the moisture-laden air within for the relatively dry, outside air. This may be done by providing cracks at the movable windows by slight openings, whereby infiltration is encouraged, or it can be done more quickly and positively by opening doors or windows about the house for a few minutes, thus effecting a rapid exchange of inside and outside air.

If a quantity of contained air at, say, 70 F, could be locked up in an occupied house, so that there was no exchange of air between inside and outside, the relative humidity of the contained air would approach the saturation condition of 100 per cent humidity because of the contribution of steam or vapor from various sources, such as steam from cooking and laundry processes and the exhalation from lungs and skin of the occupants. Also, in many cases, means of increasing the humidity are employed, such as the water pan of the hot air furnace, and various forms of humidifiers to be hung on or about steam radiators. At 100 per cent humidity, the dew-point temperature is coincident with the temperature of the air, or 70 F in the case suggested above, and moisture would probably form on the inside of all exposed surfaces, including both walls and windows, if the outside temperature was lower than 70 F.

However, no house is proof against leakage, and as a result, there is bound to be a constant exchange of air between inside and outside. Suppose the outside temperature is 10 F and its relative humidity 80 per cent. Each cubic foot of outside air contains a very much smaller amount of moisture than does each cubic foot of inside air. Consequently, every exchange results in carrying some moisture out of the house, thus maintaining a condition of air inside at a very much lower relative humidity than 100 per cent. The actual humidity maintained

will represent an equilibrium between the rate at which moisture is contributed to the inside air from the various sources, on the one hand, and the rate at which exchange takes place between the inside and outside air, on the other.

As an example, suppose the air in the room is at 70 F and 40 per cent relative humidity, while the outside air is at 10 F and 80 per cent relative humidity.

The weight of moisture per cubic foot of inside air is 0.000461 lb and the dew-point is 44 F. If there is considerable wind, the inside temperature of the glass of the windows is found to be about 25 F. The glass surface is, therefore, 19 F below the dew-point, and there will be heavy and rapid deposition of moisture, which will immediately be frozen into ice, since the surface temperature is below 32 F.

Now, let doors and windows be opened for a few minutes, and assume that three-fourths of the air inside the room or house has been replaced by outside air. This does not mean that the temperature of the freshly introduced air in the room will be as low as three-fourths the distance between 70 F and 10 F, because the great mass of walls and furniture will immediately heat the relatively slight mass of air, so that perhaps the lowest temperature registered by a thermometer might not be below 50 F.

The weight of moisture in each cubic foot of the newly introduced air will be 0.000088 lb, and the weight of the moisture in 1 cu ft of the mixture of the old air and the new in the room, in the proportions of one to three as assumed, will be 0.000181; the humidity will be about 16 per cent and the dew-point temperature 20 F which is now below the inside surface temperature of the glass. The result is that the ice will immediately begin to evaporate from the glass. The acquisition of moisture by the air by evaporation from the glass and other sources will soon raise the dew-point temperature, and evaporation from the windows will cease, and in time deposition will again take place, but, in the meantime, some of the ice coating will have been actually removed and deposition has been arrested for a period. By a few applications of this treatment, the deposition can be stopped and the ice already formed can be removed by evaporation without melting.

This theory has been put to actual test in a residence. In one experiment, the inside temperature was 75 F and the relative humidity was 60 per cent. This high humidity can be attributed to the fact that the house was newly constructed and the plaster had not yet dried out. The dew-point temperature was 59 F. The outside temperature was 40 F. Inasmuch as there was but little wind, the inside glass temperature was about 54 F. As a result, there was considerable condensate deposited because of the high humidity in spite of the very moderate outside temperature.

The doors and at least one window in each room were opened for about 3 min. About 5 min after these were closed again, the temperature of the room had reached 73 F, and in 10 min the original temperature of 75 F had been completely restored. The relative humidity was reduced by this airing from 60 per cent to 25 per cent, and the dew-point from 59 F to 35 F, which would indicate that nearly all the old air had been cleaned out and replaced by new in the 3-min airing. Under the newly established conditions, the dew-point temperature was 35 F and the glass temperature 54 F, so that condensation was no longer forming on the windows, but instead the moisture previously deposited was actually

evaporating. One hour after the airing, the moisture had all disappeared from the windows. However, after an interval of 3 hrs, light condensation began to appear again because of the moisture taken up by the air from the plaster and from the evaporation of that from the windows as well, which necessitated another airing.

On another occasion, the outside air temperature was 5 F and a strong wind was blowing. During the night, the house had been tightly closed, and in the

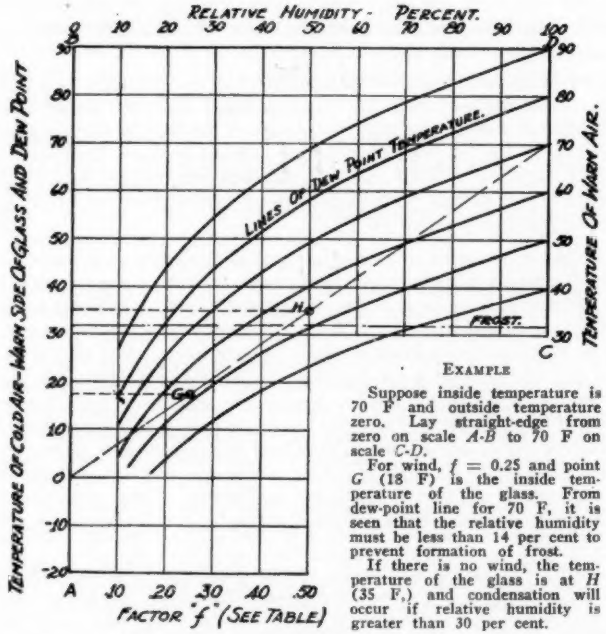

FIG. 4. CHART FOR DETERMINING RELATION OF INSIDE GLASS TEMPERATURE TO DEW-POINT AND FROST

morning, a heavy coating of frost covered nearly all the windows. In about 2½ hrs, after a single thorough airing of the house (upstairs as well as downstairs), the frost had disappeared from all but the lower panes of one bay on the windward side. In this case, the frost actually evaporated from the windows without first melting and running down the glass.

The most effective time for airing a house is just before the retiring hour at night, for it is usually at that time that the moisture content of the inside air is greatest because of the gradual increase throughout the day. During the night, with the occupants in bedrooms shut off from the rest of the house, and with other moisture sources cut off, the contained air will hardly increase enough in

moisture content to bring the dew-point high enough to permit condensation to take place.

Predetermining Frost and Condensation Conditions

As was pointed out in the earlier part of this paper, the condition that determines whether or not condensation and frost will appear upon a window is the relation of the temperature of the inside surface of the glass to the dew-point temperature of the inside air. The glass temperature is determined by the temperature of the air inside and out, by the wind and by the conductivity of the glass. The dew-point temperature of the inside air is determined by the moisture content, or relative humidity taken in conjunction with the temperature of the air.

All of these factors are presented in the chart of Fig. 4. The temperature scale on the left applies to the cold air, the glass and the dew-point; that on the right applies to the warm air. The scale of relative humidity appears at the top of the chart, and the scale for values of the factor *f* appears at the bottom. It will be remembered that this factor is the ratio of the number of degrees between cold- and warm-air temperatures. The curved diagonal lines represent different dew-point temperatures that pass through 90 F, 80 F, 70 F, 60 F, 50 F, and 40 F, respectively, at 100 per cent humidity. Thus, if the warm air temperature is 60 F (on the right-hand scale) and the relative humidity is 30 per cent (on the top scale), the dew-point temperature is 27.5 F, as read on the left-hand scale on a horizontal line through the point of intersection of the vertical 30 per cent humidity line, and the diagonally curved dew-point line that passes through 60 F on the right-hand scale.

In the example indicated on the chart, the outside-air temperature is taken as 0 F and the warm-air temperature as 70 F. A straight line is drawn from 0 F on the left-hand scale to 70 F on the right-hand scale. If ordinary single glazing is used, and there is a considerable wind outside, the factor *f* will be 0.25.

From the point where the straight line from 0 F to 70 F intersects the vertical line from $f = 0.25$, the temperature of the inside of the glass is read on the left-hand scale as 18 F. The dew-point line through 70 F warm-air temperature crosses this 18 F line at 14 per cent relative humidity. If the relative humidity is higher than this 14 per cent, for example, say 30 per cent, the dew-point temperature, 36 F, is higher than the glass temperature and condensation (which will appear at once as frost because the glass temperature is below 32 F) will form on the colder glass surface.

In the case of this example, frost and condensation can be prevented either by lowering the moisture content of the air, or by increasing the factor *f* by some means. If the moisture content is reduced so that the relative humidity becomes less than .14 per cent, no frost or condensation will appear. Or, if the value of *f* can be increased to 0.50 either by breaking off the wind, or by a fan on the warm side, then the temperature of the glass will rise to 35 F. Condensation will then occur if the relative humidity is higher than 30 per cent, but the condensation will not turn to frost because the temperature of the glass is above 32 F.

Increasing the warm-air temperature will do something towards raising the temperature of the glass. Thus, in Fig. 3, if the straight line be drawn from

0 F to 80 F, it is seen that the glass temperature is increased to 20 F for $f =$ 0.25, or to 40 F if $f = 0.50$.

Results of the Experiments

1. The thickness of the glass, within ordinary values, has but little effect upon the temperature of the inside surface. Both ¼-in. and ⅛-in. glass were tried. The thinner glass showed a slightly lower temperature, but not enough lower to warrant discrimination in the value. The resistance to heat transmission of the glass itself is so low, compared with that of the air films, that differences in thickness of ordinarily used glass introduces but a slight effect.

2. Circulation of air on the warm side of the window by a fan tends to nullify the effect of the wind on the cold side, and brings up the inside temperature of the glass to approximately the mean of the cold and warm air temperatures, which is the condition for no wind or forced circulation on both sides.

3. Insulating the glass pane from the metal member of a steel window by a felt strip or by using a framing member of wood has no effect upon the formation of condensation or frost.

4. Double glazing, consisting of two ⅛-in. panes in contact, brought the factor f up from 0.25 to 0.35 with wind. The surfaces, although in contact, appear to offer a considerable hindrance to heat transmission.

DISCUSSION

J. E. EMSWILER (WRITTEN): In Table 1 of the paper, the value of the factor f for double glass with ⅛-in. air space, is given as 0.48, which means that the room temperature of the glass is about half way between the temperature of the room air and that outside. If the coefficient of surface transmission for glass for still air is taken as 1.60 and that for outside air as three times this figure, or 4.80, and if the coefficient of the glass itself be taken as 36, then the resistance (reciprocals of the coefficients) for surface effect on warm side of window glass, two surface effects in air space, glass, and surface effect on outside of window, are respectively, 0.603, 0.028, 1.206, 0.028 and 0.201. The total resistance is the sum of these values or 2.066. The temperature drop from room air to inside surface of window should then be the ratio of 0.603 to 2.066 or 0.29 of the total temperature drop from inside to outside air. If, for example, the temperature of the inside air is 70 F and that of the outside air 0 F, the temperature of the glass on the room side should be $70 \times 0.29 = 20$ F less than that of the room air, or about 50 F. The value of the factor f as defined in the paper should be $(50 - 0) / (70 - 0)$ or 0.71. The fact that the actual value of f is only 0.48 would seem to suggest that the insulating effect of the ⅛-in. air space is considerably less than is figured on the above basis.

It is possible that with a thin space between panes, heat is transmitted by conduction rather than by convection. Using a coefficient of conductivity for air of 1.25, derived from values taken from Mark's Handbook, p. 308, the resistances for a double glass window with ⅛-in. glass and ⅛-in. air space, would be 0.603, 0.028, 0.800, 0.028, and 0.201, for warm air film, glass, air space, glass, and cold air film respectively, or a total of 1.660. The resistance of the warm air film would then be 0.603/1.660 or 0.36 of the total, and the value of f as defined in the paper would become 0.64. While the value 0.64 figured on transmission

across air space by conduction is still a good way from 0.48 as observed by the authors, it is materially better than the value 0.71 computed by assuming the passage of heat across the air space to take place by convection. It seems probable, from the standpoint of resistance to heat flow and inside temperature of the glass, that the air space of a double glazed window should be thicker than ⅛-in.

T. J. Duffield (Written): It has been so long since I have been in a house, office, or school building in which the humidity was so high that there was deposition of moisture and frost on the windows, that I did not appreciate that the matter was important and the research here reported seemed like setting up a straw man merely for the joy to be derived from bowling him over. However, there are doubtless instances both in residences and industrial plants when condensation on windows and its removal become important factors.

It is, however, of interest to know how the indoor surface temperature of glass is reduced by wind and that this reduction is not increased by velocities over 8 miles an hour. This fact is of particular importance in the light of a study of the manner in which the body loses its heat, recently made by the *Smithsonian Institution* with financial assistance of the *New York Commission on Ventilation.*[4]

When one stops to consider that at ordinary indoor temperatures, approximately 46 per cent of the heat loss of the body is effected by radiation, 30 per cent by conduction and convection, and 24 per cent by evaporation in the respiratory tract and from the skin, it will be appreciated that when outdoor temperatures are low and there is considerable wind, one may be uncomfortably cool in a room with any considerable window area, despite the fact that the air has the effective temperature determined to be the most comfortable, simply because of the heat loss by radiation from the subject to this cool window surface.

Being uncomfortably cool in an air condition determined to be most generally comfortable may seem an anomaly, but explanation is to be found in the fact that the walls of the test room in which the determinations were made were all alike and they assumed an approximately equal temperature very near that of the air, whereas a window to the outer air would have altered conditions very materially.

Paul D. Close (Written): It is interesting to note that the results of the experimental work reported in this paper check very closely with certain facts which have been available for some time and have been used to solve condensation problems.

A fundamental of heat transmission is that the temperature drop through glass or any wall or roof construction is proportional to the resistance. Thus if the total resistance of a certain construction is 10 and the overall temperature difference is 100 deg, the change through any portion of the construction having a resistance of say 1.0 would be 1/10 of the total temperature change or 10 F. The use of this fundamental makes it possible to compute the temperature gradient through any construction if the individual resistances of the component materials are known.

This principle can be readily applied to glass if the necessary heat transmission data are available. In the case of glass the so-called internal resistance is so

[4] A Study of Body Radiation, L. B. Aldrich, Smithsonian Miscellaneous Collections, Vol. 81, No. 6, December, 1928.

small that it may be disregarded and it can be assumed that the total resistance of the glass for any given set of conditions is the sum of the two surface resistances, in the case of single glass.

The air to air transmission of single pane glass according to Table 13 of THE GUIDE, 1929, is 1.13 based on a 15-mile wind velocity. The reciprocal of this quantity is 0.884 which represents the total resistance of the two surfaces for a wind exposure of 15 mph on one side. Three-fourths of this resistance or 0.663 is on the still air side and one-fourth or 0.221 on the moving air side, based on the customary assumption that the resistance of the air on the exposed side is 1/3 that of the air on the still side.

In this paper it is stated that for no wind the temperature of the inside surface of the glass is about ½ way between the temperature of the outside cold air and the inside warm air. This fact was determined experimentally. The same result can be obtained mathematically, for if the outside surface were subjected to still air conditions the two surface resistances would be equal, and since the internal resistance of the glass is negligible, the temperature of the inside surface would be the same as that of the outside surface and hence both would be the mean between the inside and outside air temperatures. The factor, *f*, derived from the experimental data can likewise be determined mathematically and would have the same value as that given in the paper, namely, 0.50 for ***no wind*** conditions on both sides of the glass.

The value of *f* of 0.25 based on wind exposure on the outside surface and calculated from the experimental data, can also be obtained mathematically. As previously stated, the resistance of the outside glass surface for wind exposure on that surface is 0.221, and since the total resistance of the glass under these conditions is 0.884, the temperature change (rise) through the outside surface would be $\frac{0.221}{0.884}$ or ¼ of the total temperature change through the glass. If the outside air temperature is 10 F and the inside air temperature is 70 F the outside surface temperature would be $10 + \frac{1}{4}(70 - 10)$ or 25 F.

Knowing the heat transmission of any type of glass, the inside surface coefficient and the inside and outside temperature conditions, it is possible to calculate the inside surface temperature of the glass, from which the maximum relative humidity which can exist in the room without condensation taking place can be determined. For example, the transmission of double glass exposed to the wind according to Table 13 of THE GUIDE, 1929, is 0.45 and if we use the still air surface coefficient of glass of 1.50 given in Table 4, we find that the temperature of the inside surface of the glass is about 50 F, based on inside and outside air temperatures of 80 F and zero, respectively. Taking 50 F as the dew-point and 80 F as the dry-bulb, the allowable relative humidity is 35.6 per cent. The value obtained from Fig. 4 of this paper is 25 per cent, the discrepancy undoubtedly being due to the difference in the widths of the air spaces in the two cases. The air space on which the tests reported in this paper were based is ⅛ in., whereas the transmission factor of 0.45 used in the above calculations was undoubtedly based on a wider air space. The wider the air space the lower the heat transmission up to a certain point and the higher the inside surface temperature for any given set of inside and outside air temperatures. Hence, a wider air space up to a certain limit will permit a higher humidity.

The statement is made that increasing the circulation of air on the warm side of the window tends to nullify the effect of the wind on the cold side. This result would be expected and can likewise be proved mathematically. Increasing the outside wind velocity, decreases both the outside and overall resistances but increases the ratio of the inside surface resistance to the total resistance which in turn decreases the inside surface temperature and hence lowers the humidity that can be carried in the building without condensation taking place on the windows.

On the other hand, if the air circulation over the inside surface is increased, the inside surface resistance is reduced, and the inside surface temperature increased, thus permitting a *higher* humidity without condensation on the windows. This latter principle is frequently used for preventing condensation or frost on store windows during cold weather. Thus, increasing the outside wind velocity increases condensation and increasing the inside air velocity decreases condensation.

This paper suggests increasing the inside air velocity and the use of double glass for preventing condensation. The objection to the former is the increase in the heat loss through the windows, resulting from the increase in air motion. From the standpoint of maintenance cost, the use of double or triple ply glass is preferable, but is objectionable as stated because of the difficulty of preventing infiltration of humid air into the spaces between the panes of glass.

E. B. LANGENBERG: We have had a number of cases of warm air heating systems where we had an excessive amount of humidity and condensation on the windows. In two cases I have been able to find the proof: one was a spring under the house which brought water up underneath the furnace. There seemed to be an attraction in the furnace caused by the air passing over the floor inside the casing which drew the moisture. To solve the problem, we had to put a galvanized iron sheet on the floor.

In the other cases—we had five—we have not been able to determine the cause. Some are frame and some brick, old houses. We studied excessive boiling of food, and even looked for a subterranean still that might be there.

One remedy is we can introduce 10 per cent outside air in which case we found the outside humidity has been very low and as the air comes in it absorbs and dissipates the humidity to a point where it does not condense on the windows.

Outside of those two things I cannot find any remedy for this condition.

W. H. CARRIER: This paper is, as the President has already said, a very important and informative one. The data are valuable for general use. I am rather curious about his difficulty with getting excessive humidity. In the East we no longer find much if any of that trouble except when the house has just been built, and some plaster is wet.

When I was a boy and lived on the farm, we had the cooking right off the living room, and we met those conditions. I don't know that these are the occasions or the particular things that they are talking about, but I have a kind of suspicion that this condition is much more healthful than the overheated and excessively dry conditions we get in our modern homes under ordinary conditions.

A part of this difficulty of getting sufficient humidity, which is much more

frequent failure than the reverse, is due perhaps to improper building construction where windows are just "thrown in" and covered with the trim instead of being caulked in or being built around properly so as to avoid loose windows and leakage. Fire-places will draw out air regardless of construction, and I believe any building that has a fire-place with a damper, as most buildings being built in the East today have, the real remedy is to open the fire-place damper.

For my part, I like to see a little condensation to know you have a little moisture there. That is the practical limit.

Then, too, it is a question of climate. The desirable humidity in a home is about 35 per cent; above that you are apt to get condensation in our milder climates around New York; below that the conditions are noticeable for dryness, excessive dryness.

I would suggest in colder climates where difficulty is found with condensation, that the real remedy is to use double sash. We hardly need it in the neighborhood of New York, Philadelphia; possibly it might be needed in Chicago; certainly all through the Northwest.

Right now I have two humidifiers delivering altogether about half a gallon of water an hour into the house, and that just barely keeps the humidity enough, and I can see just a trace of condensation on a cold day.

E. S. HALLETT: In our schools, of course, we have humidity enough to make condensation on windows on any cold day, and the effort that we have made is succeeding pretty well; that is having a diffuser in the end of the school room that throws an air draft across the window. That gives the amount of air movement that prevents excessive accumulation on the glass.

No. 834

COOLING AND HUMIDIFYING OF BUILDINGS

By S. C. Bloom,[1] Chicago, Ill.

MEMBER

THE broad contact which the public has had with the comforts of cooling in the theatre, is resulting in wide-spread interest in its possibilities of application to all types of buildings, primarily for the purpose of escape from the elements during summer.

We do not permit ourselves to be tortured by the rigors of winter, hence heating plants are a matter of course in every kind of building. It does not require the gift of prophecy to state that the time is close at hand when provision for comfort in hot weather will be classed among the necessities.

In the theatre today the cooling system is an essential feature of the equipment, for experience has shown it to be an investment yielding large returns. It is only natural that the fields in which the benefits of cooling are of a tangible nature should be the earliest to adopt it. Hence we see the hotels, stores and restaurants beginning to utilize this effective auxiliary to attract the public and provide for its comfort while within their establishments.

The greatest field on the horizon, however, lies in the cooling of office buildings. It is more difficult to evaluate the direct benefits which accrue from the maintenance of conditions under which people can work to best advantage, but it is the common experience of every one that when the weather is so hot and uncomfortable as to be distracting, there is a decided reduction in mental and manual capacity for work.

The problem of the engineer is to devise methods and means whereby the great benefits to be derived from the general usage of cooling can be attained. It is not an easy task, as any one will quickly recognize who examines the subject, yet not insurmountable. The principal obstacles at the present time are the initial and operating expense of the equipment required according to current practice, the space requirements, and the lack of a background of experience by which to determine the worth of the investment.

The last factor named is perhaps the most important, yet we have some outstanding examples of office building cooling, covering many years experience and it is the unanimous opinion of the managements that they could not afford to be without them.

The multi-story building, where population density is small, presents some interesting considerations. There is a high percentage of window area, wide vari-

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ations in the intensity and distribution of solar radiation and windage, flue effect and the necessity for conserving usable or rentable area.

The heat to be absorbed by the air supplied for cooling is preponderantly sensible—a characteristic which may prove important in the ultimate solution of the problem, through the objective of maintaining lower relative humidities than are now considered satisfactory, and higher room temperatures. It is not the heat but the humidity which is the common complaint.

At the present time, air circulating systems, where cooling is employed, are designed very much along the same lines as ordinary ventilating systems, except that the exhaust system is usually arranged for recirculating a very substantial portion of the air supplied. Existing standards, whether traditional, code or statute are, on the surface at least, adhered to.

Recirculation of air is done for the purpose of saving refrigeration, tempering the air supply and obtaining desirable humidity conditions in the building. The interesting point is that this results in a reduced intake of outdoor air. This immediately arouses speculation as to the minimum amount of outdoor air which must be introduced to a building.

There is a very considerable body of evidence which has been accumulating for the past few years from various directions, indicating that the outdoor air requirement may be lessened without seriously impairing the quality of ventilation, especially with use of ozone in odorless concentrations and the promising prospects of ionization as adjuncts.

There is, and doubtless will continue to be, resistance to acceptance of this idea, until its soundness has been more fully demonstrated. From the standpoint of cooling, its acceptance would be of tremendous significance for the reason that it would no longer be necessary to mask realities; recirculation of air, aesthetically forbidding at best, could be dispensed with. Stripped naked there would remain a simple system supplying a relatively small quantity of cold, purified, outdoor air to the spaces ventilated.

Indeed, exhaust systems may well be eliminated, thus requiring the air to find its escape through leakage and to that extent reduce infiltration of outdoor air—a factor of large consequence in air and refrigeration requirements.

If air supply can be reduced to a point which approaches the normal infiltration—a prospect which is not unreasonable, considering that occupied area will rarely be less than 30 sq ft per person—it will be seen that an important avenue of progress is opened. Taking a 10-ft story height, one air change per hour and the population density just given results in an air supply of 5 cu ft per minute per person.

The less air supplied, the colder it must be of necessity, and the lower will be the relative humidity in the ventilated spaces, and the higher the temperature for a given comfort effect. This all works in the direction of smaller conduits for the air supply system and, with the omission of a recirculating system, to a large saving in space requirements.

Consider in addition the possibilities in this direction, of increasing air velocities in the main conduits to 5 or 6 times those commonly used and of operating blowers at pressures ranging upward to 1 lb per sq in. instead of the customary range. In dealing with greatly reduced air volumes, the total power requirement for air circulation will compare favorably with the conventional systems. Fur-

thermore, we have at hand blowers which are eminently practical and suitable for such duty.

The raising of air pressures on the mains will go far to solve the problems introduced by the chimney effect in a building where many stories are interconnected by a common piping system. Similarly, the maintenance of sufficient pressure at branches will facilitate meeting the effects of windage.

It is recognized that such a piping system would require insulation, but it would not be very heavy because its principal purpose would be to eliminate external condensation, and the low dew point of the air in the building would reduce the tendency for condensation. The insulation would also muffle any noise set up by the air at the higher velocities suggested.

Undoubtedly it would be necessary to abandon the ordinary galvanized duct work, and use a form of conduit in the shape of light pipe, but there are many forms of conduit which can readily be adapted to the conditions proposed.

There remains only the question of introducing the cold air to the enclosures, in such a way as to prevent drafts. The lines along which this problem can be met are quite well known. Briefly it is through the use of very small columns or very thin streams of air, introduced in a direction and at a speed which will affect their admixture with the room air before the cooled air falls to the occupied zone.

Rough control could be affected by varying the temperature of air leaving the air cooler as outside weather conditions change, and more refined control by variation of volume of air introduced to individual rooms.

During the heating season, the same system could be used to moisten the air by the simple expedient of using a relatively small quantity of highly humidified air, the dew point of the air supplied being varied as outdoor conditions change, thus keeping the relative humidity in the rooms within a satisfactory range.

It is certain that in the general application of cooling to large buildings, we shall probably go through an experience analogous to that which accompanied the introduction of blast heating, and it would require one of great vision to suggest at this moment what the general features of the ultimately desirable system will be.

In the broad outline of the system described, the thought has been primarily to suggest some avenues of approach to a problem, upon the threshold of which we stand; a problem which demands a satisfactory solution, and requires the earnest thought of every heating and ventilating engineer. The public will have no patience with a wavering or uncertain attitude on the part of our profession.

It is not to be inferred from this warning that as a group we are not alive to the situation, for there are many, well-qualified, forward-looking engineers who are concentrating their talents upon this problem; men who are unprejudiced, who are not slaves to the conventional and who are boldly treading new and uncharted paths, confident of arrival at the goal of their endeavors.

What is needed is broad, constructive thinking and discussion of the many phases of the problem in order that errors may be reduced to a minimum. Otherwise progress in the usage of cooling will be retarded, since failures have the unfortunate characteristic of being much more conspicuous than successes.

The cooperation of the refrigerating engineer, electrical engineer and experts on municipal water supply is very important, to the end that we may not start

off on a tangent and soon find a barrier through failure to recognize the significance of some factor or factors in the engineering branches named, which precludes further advancement along the line projected.

The large power and condensing water requirements together with their load factors offer a very fertile field for investigation, especially in the way of distributing them over the period when the air cooling equipment is inoperative. Some work has been done in this field, but the results have not been uniformly satisfactory; however, enough success has been attained, in instances, to warrant further consideration.

A tendency which ought to be discouraged is the effort to maintain temperatures too low. It is not only expensive but unnecessary and even undesirable. A sane temperate view of the matter should be taken. A condition of 85 F and 40 per cent relative humidity would be unbearable in a room during the heating season, and yet be gratefully comforting when it is 95 F and 50 per cent relative humidity outside. The provision of a reasonable contrast between indoor and outdoor conditions should be the objective. Excessive cooling in summer is just as objectionable as overheating in winter.

So long as the answer to the question of what is sufficient remains in the realm of opinion, there will be some diversity of viewpoints. There is, however, some research work being done, which, with experience in existing installations, should serve as a valuable guide in determining correct practice.

A principle which may offer a basis for an indoor temperature and humidity schedule in summer, is that the necessary readjustment of the body heat controls in passing from outdoors to indoors and *vice versa* should be a minimum commensurate with suitable comfort contrasts.

If it be assumed that an effective temperature of 66 F in the environment of an average person at rest and normally clothed, is the condition wherein the body heat generated is thrown off with the greatest facility and the least discomfort, then it would seem reasonable that the maintenance of an effective temperature indoors, midway between that prevailing outdoors and 66 F, would afford the maximum comfort contrast with the minimum disturbance in the body temperature control mechanism of those passing indoors and outdoors.

On such a schedule, the contrasts would lessen as summer weather conditions recede from the extreme, until with 66 F effective temperature outdoors the contrast would vanish. There could of course be various combinations of temperature and humidity indoors to produce the required effective temperature.

It is common experience that after a protracted period of warm weather people become accustomed to it to a very considerable extent. In other words, there seems to be a seasonal adjustment of the body temperature controls, during which time we are quite conscious of discomfort. After the adjustment has been completed, the discomfort lessens or ceases. It is possible that this reaction is in part psychological, and that the cessation of discomfort results from acceptance of the idea that it is useless to quarrel with the elements.

The human body can adapt itself to variations in the condition of its atmospheric environment, throughout a reasonably wide range, without difficulty. It would seem logical then in the artificial cooling of buildings to minimize the variations to which the occupants are subjected in passing into or from a building, bearing in mind that the change in conditions is abrupt and hence their

influence is temporarily magnified. As lighter clothing is worn in summer and street dress is the same indoors, the body is more sensitive to changes in effective temperature.

When the idea of cooling buildings was first advanced, the prospective users insisted upon indoor temperatures of 70 F and 50 per cent relative humidity regardless of outdoor conditions. Usually the expense involved in securing this combination effectually stopped further consideration. There were a few such installations made and experience quickly demonstrated the futility of the idea.

The prospect of providing for the comfort, happiness, well-being and efficiency of the great body of people, compelled to work in buildings by virtue of the necessity of earning a livelihood, ought to stimulate the imagination and ingenuity of engineers and the solution of the problem will reflect great credit upon our profession.

DISCUSSION

J. S. ST. JOHN: I should like to leave a thought with the members of this Society on a subject that was brought before the *Society of Automotive Engineers* recently—automobile body heating and ventilating. The automobile men admitted their utter failure to secure something that was good and definite in that matter. A most vital example they gave was in a blinding rainstorm on a summer day, when the humidity was bad, the temperature around 80 F, all the windows and ventilators of the automobile were closed; therefore, no ventilation of any kind whatsoever.

Every method of automobile ventilation depends entirely upon convection, the movement of the automobile, and automotive engineers are putting forth a program during this coming year, that they are going to work intensively on this subject. This may be a thought for the members of this Society to think of in their leisure time and also show the interest other branches of industry show in one of our most important subjects, cooling and humidifying.

E. S. HALLETT: We think of school engineers as a conservative portion of our membership for schools generally, I suppose, are following rather than leading, as compared to theatres and buildings of that kind, yet right at the present moment, in St. Louis, we are designing an auditorium which is to be cooled just as the great successful theatres of our city are cooled. We are going further than that. We are contemplating very soon, within a year or two, a new office building for the Board of Education, and I was very much interested in the thought that was offered in this paper on the cooling of an office building, as it applied to our new office building.

We have another feature in that there are to be no radiators in this building, concealed or otherwise, as the building is to be heated and ventilated during the winter season entirely by the blast system exactly as we take care of our schools.

That may be a novelty to a good many of you, but we have our plans well laid for accomplishing that result, and we contemplate in this connection to use the same system, so far as possible, in keeping all year round comfortable temperatures in this building.

I want to call attention to the cost of lack of cooling in our schools. Last year we had 10 days in which it was impossible to conduct school on account of high

temperature. Ten days loss means one-twentieth of the entire school year lost. One-twentieth of $14,000,000 represents the cost of lack of cooling for the school system.

Now, it may be in the quite remote future that we can reach that. I am using this merely as an illustration of the cost of it, not that we are going to be able to cool our schools immediately so as to be comfortable.

There is another period, other than that in which we close our schools, when the temperature reaches 90 F outside, but after the temperature reaches 80 F outside, might just as well be closed, on account of the discomfort, and that is a matter that must be considered.

If the theatres must be absolutely empty, as they would in a city like St. Louis, during the summer season, for three months without the cooling system, what may we expect, then, of other buildings in the city?

As an illustration one of our large concerns, the Graham Paper Co., had been having difficulty in securing office output, and they had a forward-looking man in charge of their office, who came to me and said, "What do you think about cooling that office building?"

I said, "It will probably add to the efficiency of your office 10 per cent, add that much to your working force of 300 employees. In other words, you may turn off 10 per cent of your employees and still get as much work done as you have done." He installed the equipment, and he told me afterwards that they did better than he had expected. He was delighted with it.

The next few years this Society will be hearing paper after paper of this kind, and I feel very sure that the time will soon come when we shall recognize that it is just as necessary to have cooling in the summer time as it is heating in the winter. We never lose a life in cold weather. Nobody ever freezes to death. Nobody dies from the exposures of cold in these days, and yet in St. Louis, there is never a summer passes that there aren't quite a number of persons who die from high temperature.

Those are all things that would indicate the truth of the paper that has been read before us, and I hope that when we go back to our homes that we shall plan to put into actual operation the building and equipment and necessary machinery to accomplish this result.

M. C. W. TOMLINSON: I did not get here in time to hear what the speaker had to say, but I think I can say something to the Society along the line of revolutionary things that are about to happen in air conditioning that may sound rather odd and rather revolutionary.

For the last three years in the electrical industry we have had some experience in using very low humidities, lower humidities probably than have been used commercially before. We have gone as low as a half per cent. We now know what that humidity will do in the line of protecting insulation, after it has been dried by the best drying processes, until some method of sealing or impregnation that will keep moisture out can be applied. We get very much better electrical characteristics than have been possible before.

In addition to this, I had noticed for quite a while that laborers or truckers, who were trucking heavy material in and out of this type of room (a room that has been air conditioned at very low humidities) had no record whatsoever of absences due to colds or pulmonary troubles. It struck me that the time was

going to come when we would do a type of air conditioning very similar to that in our hospitals and sanitariums.

Out in Arizona, where the relative humidities are around 10 per cent, and the temperatures are quite high, is where we send people who have tuberculosis. Why cannot a man who has the means to afford that sort of thing be treated right at home in that type of a room in a nearby hospital? In that way he could keep in touch with his business. This was the thought that ran through my mind. That has not been done yet but there is no reason why it cannot be done. As a proof that it can be done there is a hospital in New York City where they have been trying out air conditioning for pneumonia. You will probably be surprised to know that they started out at about 40 per cent relative humidity with a temperature of about 70. It was not very long before they conditioned this room down to 30 per cent.

It has only been about a week ago since I was talking to the specialist who has made this venture and I told him that I believed it was not going to be a great while before he would be working at very much lower relative humidities. He asked why, I told him why as I have told you and he said, "I know you are right. We can't, with the installation we have, go down lower than 30 per cent. I have been wanting to get below 30. I don't know where we are going to stop but we must have lower relative humidities in order to obtain better conditions for these patients."

I think that he is going to go down to at least 20 per cent and I wouldn't be a bit surprised if they would finally wind up around 10 per cent relative humidity. In air conditioning we usually start very much higher and go lower and lower as capacity of our equipment will allow.

Here is another thought: The work that has been done along the line of human comfort is very little known. There are other professions that ought to know a whole lot more about it than we know. This specialist knew very little about it. Yet he had been in touch with a consulting service. I also told him, that I thought his temperatures would have to go up, as his relative humidities went down.

I have seen with my own eyes human beings lying in that room who should have passed out by all methods of figuring by the medical profession of today This is a very wonderful venture in air conditioning, it seems to me, when you can save human lives.

I hope we will find some way of applying air conditioning through which we will not only be able to cut down our pay rolls, by reducing our employment, as the previous speaker has suggested but also we may find some means of using it for increasing employment.

There is one other thing that I might say and that I feel I can say at this time. That is, that with relative humidities around 5 per cent, air motion of 100 fpm and temperature around 85 F, it is very chilly. At relative humidities about a half per cent with the same air motion and temperature it is cold. Also it is comfortable.

J. I. LYLE: I want to congratulate the Society on this splendid paper of Mr. Bloom's. There is a lot of real thought in this paper. I would just like to emphasize one or two things that Mr. Bloom brought out. He spoke of the theatre owners and the early work that was done in cooling the building. Unfortunately, the theatre owners and operators are artists. They know about the box

office and they know about the stage or the screen, but they know very little about human comfort. They are advertisers; they want to impress the public with the fact that they have a cooling system, and as a result they have greatly injured the cooling industry because people have reached the point of thinking that they are going to have pneumonia every time they go into a cool theatre.

It is not a question of what is the most comfortable effective temperature. It is a question more of its relation to the outside condition. If we will go into a cooled room and stay for some time we may find a very much lower effective temperature is comfortable than is anything like comfortable when we first enter. That is, in cooled places, we have to watch for shocks, both as we enter the building and as we leave it.

Usually in warm weather, there is more or less perspiration carried by our clothes; we go into a room with a lower humidity than that existing outside, and with somewhat lower temperature, and there is immediately a very rapid evaporation from our clothes which gives a very unpleasant chilling effect; and then after we have become acclimated to the internal conditions and we go outside, it is just like being hit in the face with a bottle of hot water or something of that sort because of the shock of the high temperature and the high humidity from outside. There is a factor that we have to take into account, and one in which I do not believe any real determinations have yet been made as to what we should have. I am quite sure that we should have effective temperatures higher than the effective temperatures of maximum comfort for summer time conditions.

Then Mr. Bloom has brought out the question of using high pressures. Some work has been done along that line. There are quite a number of installations, not to the extreme yet, that he has suggested, but using pressures of several inches, water pressure, instead of 1 lb as he suggests, using a very small amount of air per occupant, and those installations are probably among the most successful that have been installed. That kind of system for the type of building that he was discussing, a high office building, is quite essential because of the size of ducts reduced to the point where you can install them, but more than that is the fact that such a system lends itself so much better to control—I mean to distribution and the regulation of the amount of air that is to be given to each room.

The ordinary ventilating system on a high building of that type may be perfectly balanced and somebody opens the window and two or three other people shut off the air supply, or something of that sort, and it is all out of balance. With the higher pressures it is not so easily disturbed.

It was an awful indictment that Brother Hallett brought on St. Louis, of all the deaths they have there from high temperature. I was surprised that he would admit that St. Louis had any shortcomings.

He spoke of ten days' loss of school, actual shutdown. If he made a study of it, I will prophesy he will find the ten days' loss was a small percentage of the loss of close application, of real work, on the part of the children in the school on the days when they did operate, but when the conditions were such that the children were nervous and fidgety and didn't apply themselves to the work.

Now, we have found that in offices, in factories, various places, where the efficiency, even on days when you could operate, has been so greatly increased that

it would readily pay for the addition in cost of a cooling system over that of what an ordinary ventilating system would require.

Referring to Mr. Tomlinson's remarks, I might say that this question of having proper conditions at home has received a great deal of study by many engineers. There are equipments now being built that not only control the temperature and humidity and air motion, but in which the pressure within the rooms can be regulated from a 20-in. barometer to a 40-in. barometer, with the humidity and temperature regulated and in which ultra-violet rays or red spectrum rays are applied.

DR. E. V. HILL: This is certainly a fine paper and a very interesting discussion. There are some points in it that I wish to emphasize, although I do not feel that I can add anything of material value. In the first place, I hope you will pardon me if I call attention once more to the fact that no one, so far as I know, has used the term "fresh air" in any of these discussions.

You still laugh at that. This is the fifth year I have been talking about it, and we have pretty well put the message across. When you quit talking about fresh air, when you quit thinking about the wonderful condition of nature and how futile are the efforts of man, then you begin to get somewhere with air conditioning.

Now, the point in this paper that struck me at once was the suggestion made by Mr. Bloom that you introduce enough air from the outside to take care of the leakage into the building. When you begin to do that, you have forgotten that you need fresh air, and I think that his position is 100 per cent sound.

Another thing I want to speak about is that we heard a paper yesterday that blazes a new trail in heating. At the last meeting of the Illinois Chapter, Mr. Hartman spoke on ionization, all of which impresses me with the wonderful awakening of this Society to the great possibilities that lie before us. Forget that the outdoors is the ultimate desirable condition; realize that we can make conditions inside a whole lot better and a whole lot more healthful.

Some years ago, when I was in the city service, Dr. Robertson was Chairman of the Municipal Tuberculosis Sanitarium, and I suggested to him that we could construct a sanitarium for the treatment of tuberculosis with air conditioning that would be far superior to any haphazard conditions we might find out of doors, and it would be in effect 24 hours a day, 365 days in the year.

He was convinced of that, and began the necessary maneuvers to get an appropriation to build an air conditioned hospital for the treatment of tuberculosis. Unfortunately, the political situation changed about that time, and it was lost in the shuffle. However, I think, as one gentleman just remarked, that is the next thing.

One thing more I want to say; I was down to St. Louis a few days ago with Mr. Hallett, making some tests in some of the St. Louis schools, and I was impressed again with the excellent air conditioning in those schools, where he is recirculating I guess 90 or 95 per cent of the air, and ionizing it. He says ozonizing, maybe it is, maybe it is ionizing; I don't know what it is. A good many in this Society following the lead of Jordan and some of the great lights in chemistry, who have rather turned up their noses at ozone.

However, we have to appreciate the wonderful possibilities, of ionization or whatever it is, but let's get into the habit of thinking that we can make air

conditions better indoors than we have outdoors. I am firmly convinced that is the truth.

There are lots of reasons for so believing. Take the general health, the stature, the intelligence of the people living in the city today, and it is far superior to that of the people in the country districts. Of course, there are other reasons. Our problem—and it is not so far off in solution—is to improve upon what has gone before.

S. C. BLOOM: I cannot add anything to the discussion. It is a great satisfaction if this paper has stimulated thought on the problems which face us in the application of cooling to buildings generally.

I want to say that I feel that we are going to be face to face with the necessities of that situation perhaps sooner than most of us think, and we do not want to be caught unaware. A great many of us are giving some thought to this and if we would design in our minds the kind of cooling system that each of us would apply, out of the mass there should come some good ideas.

As to Dr. Hill and his remarks about "fresh air"—I am willing to admit that in preparing this paper every time I came across a sentence where the term might be used and I was tempted to use "fresh air," I did not use it.

No. 835

SEMI-ANNUAL MEETING, 1929

FOR the first time in the history of the A. S. H. & V. E. a meeting was held outside of the U. S. A., when the semi-annual gathering was entertained by the Society's Ontario Chapter at Bigwin Inn, Lake-of-Bays, Ontario, Canada, with 380 present.

Definite forward progress was made in the solution of the boiler testing and rating problem, a code for testing and rating unit heaters is in prospect, and regulations for the heating and ventilating of garages were adopted so that the Semi-Annual Meeting 1929 can be put on record as one of progressive action.

The Semi-Annual Meeting of the Society was called to order by President Lewis on Wednesday, June 26, and M. Barry Watson, president of the Ontario Chapter, welcomed the members to Canada for the first meeting of the Society, outside of the United States. In response, President Lewis voiced the appreciation of the Society for the splendid work of the Ontario Chapter members and said it was fitting that a scientific Society should not be restrained by national borders. President Lewis said that science is international and men engaged in the same profession are brothers no matter where found.

D. E. French reporting for the Committee on Code for Testing and Rating Unit Heaters which had been working jointly with a Committee of the *Industrial Unit Heater Association* in the preparation of this Code, stated that the joint committee had prepared a Code for Testing and Rating Steam Unit Heaters and it had been presented to the *Industrial Unit Heater Association* on June 25 with the result that the association accepted the report of the joint Code Committee, with certain changes agreed to by the Committee, as an amendment to the rules for testing and rating which the association had previously adopted. The Association recommended that all member companies put the prescribed test procedure in use for trial and suggested that the present committee be continued so that it might present the Code for adoption at the *Industrial Unit Heater Association's* meeting next January and revised to include any improvements that might result from the experience of members who use it.

Mr. French pointed out that in view of the action of the *Industrial Unit Heater Association* in adopting the report of the joint Code Committee as a tentative standard, the joint committee desired to report this action to the Society and felt that it would be ready to present a Standard Code to the members at the January 1930 meeting.

Report of Endowment Fund Committee

The Secretary, A. V. Hutchinson, read the report of the Special Council Committee relating to a proposed Society Endowment Fund which had been prepared and sent out to all Society members prior to the Summer Meeting.

May 24th, 1929.

TO THE MEMBERS OF THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS:

At the Semi-Annual Meeting of the Society, June 26, at Bigwin Inn, Lake-of-Bays, Ontario, Canada, the report of a Special Committee of the Council, relating to the creation of a Society Endowment Fund, will be presented for the approval of the A.S.H.V.E. members.

The Committee, A. C. Willard, Chairman, John Howatt and W. T. Jones, was appointed by resolution of the Council March 8, 1929, and the Council received the report of the Committee and approved it at the May 13, 1929 Meeting.

The report of the Committee and the resolutions of the Council follow:

THE SOCIETY ENDOWMENT FUND

(1) In order to conserve certain specified profits accruing to the Society from certain of its agreements and activities, it is proposed to establish a Society Endowment Fund[1] to be administered solely by the Council as a Depository for the net profits from the agreement with Domestic Engineering Publications dated January 10, 1929, and such other balances, receipts, donations and bequests as may be received from time to time and specifically designated by the Council of the Society to be paid into the Society Endowment Fund.

(2) This Fund is to remain for all time a permanent endowment fund, and only the income from the Fund may be devoted to such activities, purposes or uses of the Society as the Council may approve by a favorable formal vote of not less than three-quarters of the entire Council membership.

(3) The income from the Fund remitting from the agreement with Domestic Engineering Publications is to be restricted entirely to the research activities of the Society and can be applied to no other purpose except by formal action of the Council in the manner already indicated in paragraph (2).

(4) This proposal is to be sent by mail before June 1, 1929, to the entire membership of the Society as a recommendation of the Council, and is to be voted on at the June 1929 meeting of the Society.

April 30, 1929.

Submitted by the
COMMITTEE ON ENDOWMENT FUND
John Howatt,
W. T. Jones,
A. C. Willard, *Chairman.*

On motion of Mr. Langenberg, seconded by Mr. Carrier, it was

VOTED: That the report of the special committee be accepted and that the Trustees to administer the fund be the President of the Society, the Treasurer and the Chairman of the Finance Committee.

and it was further

VOTED: That this report be placed before the Semi-Annual Meeting, 1929, with the Council recommendation of approval.

THORNTON LEWIS, President. A. V. HUTCHINSON, Secretary.

[1] This new fund is not to be confused in any way with the present Research Laboratory Endowment Fund, which is entirely separate and distinct from the new Society Endowment Fund.

President Lewis stated that the report was before the meeting for discussion and called attention to the action taken by the Council.

Summarizing the points which had been considered by the Council, President Lewis stated that it was felt unwise to designate the income for use entirely for Research Laboratory activities and for that reason, it was thought desirable that an Endowment Fund be set-up leaving it to future Councils to appropriate the income for any purpose which might be desirable. While it is the Council's function to appropriate moneys belonging to the Society for any purpose, the members felt it advisable to get the advice of the Society to find out the wishes of the members. The Council will obtain legal advice when it formally acts in this matter.

After a lengthy discussion, W. H. Driscoll said that the resolution was laudable in its purpose but it would be questionable for the Society to go on record in favor of the resolution as drawn or on a substitute resolution with any idea of summary action in disposing of the matter. The legal aspects of the problem should be seriously considered. He suggested that the matter of establishing an endowment fund be referred back to the Council so that if in its wisdom and judgment of the Council determines that it is advisable to establish a fund it could have definite recommendations available for members prior to the Philadelphia Meeting and take action in January.

This was made as a motion and seconded by John Cassell and unanimously passed.

L. A. Harding, Buffalo, reported as Chairman of the Continuing Committee on Codes for Testing and Rating Steam Heating Solid Fuel Boilers and said that three definite items would be brought before the meeting.

First the Committee requested approval of some slight changes in the A. S. H. & V. E. Standard and Short Form Heat Balance Code for Testing Low Pressure Steam Heating Boilers, designated as Codes 1 and 2. The changes recommended by the Committee were the definitions of the term "grate area" and of the factor of evaporation.

Mr. Harding explained that, in view of the Society's expressed attitude relative to the rating of boilers on the basis of tests, it was necessary to have a satisfactory performance code and as the heat balance codes are not well adapted for rating tests, a little different form designated as A. S. H. & V. E. Performance Code for Testing Solid Fuel Boilers (Code No. 3) was submitted. This Code he said is identical except in one or two minor cases with the Standard Boiler Code recently adopted by the *National Boiler and Radiator Manufacturers' Association*.

The reason for the slight changes made by the Society is to put this Code in conformity with the values used in the Code of Minimum Requirements for the Heating and Ventilation of Buildings, where the latent heat of evaporation is given as 971.7 as compared with the value used by the manufacturers of 970.4.

On motion of R. V. Frost, properly seconded, the revisions of Codes 1 and 2 were unanimously adopted.

Motion for the adoption of Code 3 was made and seconded and unanimously carried.

STANDARD AND SHORT FORM HEAT BALANCE CODES FOR TESTING LOW-PRESSURE STEAM HEATING SOLID FUEL BOILERS

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

Codes 1 & 2

REVISION OF JUNE, 1929

I. INTRODUCTION

THE purpose of the Standard and Short Form Heat Balance Codes for Testing Low-Pressure Steam-Heating Boilers is to provide a standard method for conducting and reporting tests to determine the heat efficiency and performance characteristics. The Code recognizes that tests of boilers may be made under different conditions, for different purposes, and with complete or limited facilities for conducting the test. It is designed to cover the determination of a complete heat balance; if a less complete test is required to satisfy the objectives of the test, observations not required can be omitted.

Two codes for recording data and results are included:

Code 1, *Standard Form, Solid Fuels,* is arranged for recording the full data called for by this Code.

Code 2, *Short-Test Form, Solid Fuels,* is for tests in which a complete heat balance is not desired; the following determinations are not required.

Proximate analysis of fuels, except as required for record, or for new-fire method.

Ultimate analysis of fuel.

Complete flue gas analysis; CO_2 is required for record.

Determination of combustible in refuse may be omitted if loss to ashpit is not required.

II. DEFINITIONS

(*a*) *Grate Area.*—The purpose of defining grate area is to fix the area which shall be used when estimating the rate of burning the fuel. The grate area shall be expressed in square feet and shall be construed to mean the area of the grate surface measured in the plane of the top surface of grate. The grate area of special furnaces, such as those having magazine feed, or special shapes, shall be the mean area of the active part of the fuel bed taken perpendicular to the path of the gases through it. For furnaces having a secondary grate, such as those in double-grate down-draft boilers, the effective area shall be taken as the area of the upper grate plus one-eighth of the area of the lower grate, both areas being estimated as defined above.

(*b*) *Combustion Rate.*—Average combustion rate per square foot per hour of a solid fuel for a given period of operation shall be the weight of fuel burned during that period divided by the product of the grate area in square feet and the number of hours.

III. SET-UP OF BOILER TO BE TESTED

(*a*) *The Boiler.*—If the boiler is set up for the purpose of testing, it shall be assembled according to manufacturers' directions.

(*b*) *Boilers* already installed shall be arranged to comply with the following conditions as nearly as possible, or as shall be agreed to by all interested parties.

(*c*) *Boiler Covering.*—The boiler should be covered with such heat insulation as is recommended by the manufacturer; if tested uninsulated or party insulated an allowance shall be made for that portion of the heat lost from the bare iron steam surface which might have been saved by covering; the allowance shall be computed from the formula given with the tables of this Code.

(*d*) *Steam Outlet Piping.*—All steam outlet pipes from the boiler shall always be well covered, up to and including the steam separator.

The vertical outlets from the boiler leading to the separator shall be in number, arrangement and size according to manufacturers' directions and shall not rise vertically from the boiler nozzle more than 5 ft before entering the steam separator.

The steam piping leading from the separator shall have a slope so that water condensed in it will drain away from the separator. The drain pipe from the steam separator shall include a U-leg sufficiently long to prevent the steam blowing out, and with a short inverted U-leg at the outlet end so that the drip may be caught in a container.

The steam valve on the boiler steam outlet piping shall be placed beyond the steam separator.

(*e*) *Feed Water Piping.*—The feed water pipe should connect to the boiler where the returns would normally come. The temperature of the feed water shall be read from a thermometer inserted in a cup projecting well into the feed line near the boiler and filled with a heavy oil. A valve shall be placed on each side of the above-mentioned cup. The valve between the cup and boiler shall serve as a shut-off valve. The valve between the cup and feed water supply shall be used as a regulating valve only. It is recommended that this valve be an automatic one for maintaining a uniform water level in the boiler. If only one valve is used, it shall be placed between the cup and the boiler.

All boiler-water connections, including blow-off pipes, must be exposed to view, so that leakage may be observed and either stopped or measured.

(*f*) *Water Gage Marking.*—The water gage shall have a permanent mark to indicate the water level recommended by the manufacturer; if there is no permanent mark, for the purposes of these tests, a mark corresponding to the level recommended by the manufacturer shall be put on the water column gage glass.

(*g*) *Smokehood Connections.*—The boiler shall be connected with a short, direct smoke-pipe to a chimney flue of suitable size, height and construction to give proper draft. All joints shall be thoroughly sealed and maintained so during the tests.

The smokehood and the smoke flue shall be insulated with not less than 1 in. thickness of insulation; the smoke-flue insulation shall extend at least one equivalent flue diameter beyond the point where the flue-gas temperature is measured.

There shall be no check draft damper between the boiler and the point where the flue-gas sample is taken or the flue-gas temperature measured; if one is incorporated in the boiler it shall be thoroughly sealed during all tests.

The choke damper, if incorporated in the boiler, shall be put in place, opened wide and kept so during all tests. A choke damper shall be placed in the smoke pipe between the chimney and the point where flue-gas measurements are made. This shall be called the regulating damper. A check damper may be placed between this damper and the chimney if desired.

(*h*) *Chimney.*—This code allows that the draft may be produced by a chimney or by a fan or other arrangement for providing induced draft; there shall, however, be a suitable damper or by-pass in the test flue or chimney that will permit of the draft at the regulating stack damper being reduced to a little above that required for the output of the test which is being made.

(*i*) *Cleaning of Boiler.*—The water spaces of the boiler shall be thoroughly boiled out with a solution of sal soda, caustic potash or caustic soda and then thoroughly rinsed with clean water.

The heating surface, firebox, ashpit, flues and chimney shall be clean and free from soot, ashes and dust at the beginning of the test.

IV. Instruments and Measuring Apparatus

(*a*) *Steam Output.*—The weight of water evaporated may be determined either by condensing the steam and weighing the condensate, designated as Method *A*, or by weighing the water fed to the boiler, designated as Method *B*.

Method A.—Water shall be fed to the boiler without being weighed. The steam after passing through the separator shall be led to a condenser through a connecting steam pipe sloping toward the condenser. The condenser shall be

so set up that all the condensate will drain quickly to the outlet. The condensate shall be caught in suitable tanks and weighed.

The condenser shall be the closed type, preferably with outside joints, and of sufficient capacity that the temperature of the condensate shall not exceed 150 F. Before each test it shall be tested for freedom from leakage by putting it under full water pressure.

Method B.—The weight of water fed to the boiler shall be determined in an accurate and suitable manner. The most convenient weighing arrangement depends on the size of the boiler, but it shall be such that the weight fed can be determined at any instant. The use of tanks calibrated for the weight of water they hold at the temperature of the feed is permissible.

The water may be fed to the boiler by gravity, air pressure or feed pumps; there shall be no leakage between the measuring tanks and boiler.

(*b*) *Weighing Scales.*—Accurate scales of suitable size shall be provided for weighing separator water, feed water, fuel and all refuse removed from grate and ashpit.

(*c*) *Draft.*—Three draft gages shall be provided and so arranged as to determine the pressure difference between the room atmosphere and the ashpit, the firebox and the smokehood. Draft measurements shall be made with draft gages reading to 0.01 in. Draft gages shall be checked for zero reading each hour.

The smokehood draft shall be measured at least one pipe diameter before the regulating stack damper. The connection into the stack flue shall have its end square and shall be set at right angles to the flow of the gases. A one-eighth in. iron pipe is a convenient size.

(*d*) *Steam Separator.*—The steam separator shall have sufficient volume to catch slugs of water and to prevent their passing into the steam line; it shall also efficiently separate entrained water from the steam carrying it.

(*e*) *Temperature Measurement.*—Accurately calibrated instruments shall be provided for measuring temperatures of gases, water and steam.

A mercury thermometer is preferable for measuring feed water temperature. It is recommended that thermocouples, in preference to other types of thermometers, be used to measure the flue-gas temperatures. The temperature shall be measured in the smoke pipe not less than one pipe diameter beyond the smoke-pipe collar. The thermo-junction or bulb of the thermometer shall be placed at the center of the pipe. The instrument used with the couple should be sensitive to at least 5 F and have a range up to 1200 F.

(*f*) *Gas Analysis.*—The sample for gas analysis shall be taken from the smoke pipe at the point at which the temperature is measured in such a manner as to give a fair average sample of the gas stream. An orsat or equivalent gas analyzer shall be used.

An open end ⅛ in. iron pipe reaching ¼ way across the flue pipe is usually satisfactory but the probable mixing of the gases should be considered and the open end so placed as to be in the average gas stream. For gas temperatures of over 750 F the portion of the collecting pipe extending into the flue should be clay or silica because the iron at this temperature may reduce CO_2 to CO.

As an average value for the test period is desired, the correct and simplest method is to collect samples of the flue gas at a constant rate into a bottle, and to take the samples for analysis from these bottles. Each of such samples shall be collected for the same length of time and their analyses averaged.

If recording gas analyzing instruments are provided, they shall be checked every hour with the orsat or other manual apparatus.

(*g*) *Smoke Readings.*—Smoke readings shall be made by the Ringelmann Chart Method.

(*h*) *Pressure Readings.*—A calibrated steam gage or a mercury column shall be used for determining the steam pressure.

V. FUEL SAMPLING

(*a*) *At regular intervals,* during the progress of the test, fair samples of the fuel shall be taken with a shovel, stored in a covered vessel in a cool place and after crushing and quartering, two one-pint glass jars or other airtight vessels shall be filled. The small samples shall be preserved for determinations of the proximate analysis, ultimate analysis and calorific value. For small boilers the total sample for each test shall be not less than 100 lb, otherwise the sample shall be approximately 10 per cent of the fuel fired.

(*b*) *After it has been weighed,* the refuse taken from the ashpit and grate shall be reduced by crushing and quartering to a quantity sufficient to fill two one-pint jars or other airtight vessels for determining its combustible content in the laboratory. Care must be taken to crush and quarter the coal, ash and refuse on a clean floor. Precaution should be taken to see that the ash and refuse does not burn after removal from the grates or ashpit.

For large boilers which are cleaned down to the grates, the large clinker masses may be separated from the other cleanings and not included in the crushing and quartering process for obtaining a sample. The respective weights shall be determined and recorded.

VI. DURATION OF TESTS

(*a*) *The duration of tests* shall be based on the firing of the minimum quantity of fuel to make the quantitative errors comparable for all tests. Every test shall include at least one complete cleaning of the fuel bed.

(*b*) *Tests using the new fire method* shall include at least two complete firing periods and the firing of at least 100 lb of fuel, excluding the kindling, for each square foot of grate area.

(*c*) *Tests using the continuous firing method* shall be run until at least 140 lb of fuel has been fired for each square foot of grate area.

VII. NEW FIRE METHOD OF STARTING AND STOPPING TESTS

(*a*) *The new fire method* of starting and stopping tests may be used on any boiler having not over 8 sq ft of grate area when burning anthracite or coke.

(*b*) *A preliminary fire* shall be made and the boiler operated under test condition for at least one hour before starting the test. The preliminary fire shall then be dumped and the ashpit thoroughly cleaned. A weighed quantity of dry wood shall be placed on the grate and kindled. The wood shall be considered as having a heating value of 5000 Btu per pound. The test shall be considered started at the time of lighting the fire. The total weight of fuel to form the complete first firing shall be charged in two portions, one-third onto the kindling wood, and the remainder within 30 min after the start of the test.

(*c*) *Before the test* is ended the fire shall be thoroughly cleaned and all ash and refuse shall be taken from the ashpit, weighed, sampled and analyzed for combustible.

(*d*) *At the end of the test* the fire shall be dumped. The residual fire when dumped shall be promptly placed in tightly covered cans, weighed and left to cool. After cooling it shall be sampled and the sample sent for a determination of its heat value. The total fuel fired shall be taken as the total weight of fuel exclusive of the wood to which shall be added the fuel equivalent of the wood and from which shall be subtracted the fuel equivalent of the residual fire.

(*e*) *The height of water line* in the gage glass shall be recorded after the residual fire is dumped. If this height differs from that recorded after the dumping of the preliminary fire, the weight of water corresponding to this difference in height shall be determined and the weight of water fed or evaporated corrected for that amount.

(*f*) *The method of recording* the data and the computations for the procedure specified in the above are given under the new fire section of Table 1, Standard Form, of this Code. An alternative procedure is the same as previously outlined except that the residual fire is only weighed and no calorific determination is required. It requires only the calorific value and the proximate analysis of the fuel. The method of recording the data and the computations are given under the new fire section of Table 2, Short-Test Form.

VIII. Continuous Firing Method of Starting and Stopping Tests

(*a*) *A preliminary fire* shall be made and the boiler operated under test conditions not less than one hour before starting the test. The fire shall then be burned low, thoroughly cleaned and the remaining live fuel spread evenly over the grate as the foundation for the first test fuel charge. The thickness of the fuel bed and the extent to which it has been burned through shall be quickly measured and recorded. The height of water line in gage glass shall be noted and recorded. The test shall start at the time of making these observations. A weighed charge of fuel shall be immediately fired and recorded. The ashpit shall be cleaned and the cleanings discarded.

(*b*) *At the end of the test* the fire shall be burned low and cleaned so as to leave the same amount of live fuel on the grate as at the start. When this condition is reached and the water level in the boiler is at the same height as at the start, record the time, and this time shall be the time of stopping. The contents of the ashpit shall be removed promptly on stopping, and placed in airtight cans, weighed and left to cool. The boiler shall be charged with all fuel fired during the test.

IX. Method of Conducting Tests

(*a*) *The steam pressure* shall be maintained at 2 lb unless the purpose of the test requires operation at some other definite pressure. The water level in the boiler shall be maintained constant during the test.

(*b*) *Observations.*—Weight of fuel and times of firing shall be recorded. The records of water fed or condensed shall be made as required by the weighing system.

It is desirable to determine the weight of water fed or condensed each hour to check the rate of steaming.

Observations of drafts, pressures and temperatures shall be made regularly during the test at least every 30 min. Separator water shall be weighed every hour.

Flue-gas analyses shall be made on samples collected for 30-min periods for high and for one-hour periods for low rates of steaming.

Smoke readings shall be made at least every 30 seconds over a period of time sufficient to show the performance under all typical conditions.

(*c*) *Fuel Charge.*—The weight of fuel charged at each firing, and the method of firing, shall conform to the manufacturer's instructions or to agreement between interested parties.

(*d*) *Attention to Fire.*—Attention to fire shall be considered as any one of the following: Firing, Stoking, Leveling, Shaking Grates, Cleaning Fire. Several of these operations performed at the same time shall be treated as one attention.

Code No. 1

Standard Form

TABLE 1. DATA AND RESULTS OF TEST—SOLID FUELS

General Information

1. Number of Test ....
2. Date of Test ....
3. Maker and Catalog Designation of Boiler ....
4. Grate Surface ....sq ft
5. Duration of Test ....hour

Fuel

6. Kind ....
7. Size ....

Proximate Analysis, as fired:

8. Moisture ....per cent
9. Volatile Matter ....per cent
10. Fixed Carbon ....per cent
11. Ash ....per cent

Ultimate Analysis, as fired:

12. Hydrogen .... per cent
13. Carbon .... per cent
14. Nitrogen .... per cent
15. Oxygen .... per cent
16. Sulphur .... per cent
17. Ash .... per cent
18. Heating Value per Pound, as Fired........ Btu
19. Total Fuel Fired during Test .... lb
20. Average Fuel per Firing .... lb
21. Average Interval between Firings .... hour
22. Average Interval between Attention to Fire........ hour

ASH AND REFUSE

23. Total Weight of Ash and Refuse Removed........ lb
24. Combustible in Ash and Refuse .... per cent
25. Carbon Burned per Pound of Fuel as Fired .... lb

DRAFT

26. Draft in Ashpit .... in. water
27. Draft in Furnace .... in. water
28. Draft at Boiler Outlet........ in. water

FLUE GASES

29. Carbon Dioxide .... per cent
30. Oxygen .... per cent
31. Carbon Monoxide .... per cent
32. Nitrogen (by difference) .... per cent
33. Temperature of Flue Gases Leaving Boiler........ F
34. Temperature of Air for Combustion........ F
35. Dry Flue Gas per Pound of Fuel, as Fired........ lb

STEAM AND FEED WATER

36. Steam Pressure by Gage, Boiler .... lb per sq in.
37. Temperature of Feed Water Entering Boiler........ F
38. Total Weight of Water Fed to Boiler........ lb
39. Total Weight of Water from Separator........ lb
40. Percentage of Moisture in Steam........ per cent
41. Total Water Evaporated .... lb
42. Factor of Evaporation ....

HOURLY RATES

43. Fuel Burned per Sq Ft of Grate per Hour as Fired........ lb
44. Equivalent Rate of Combustion of 12,500 Btu Fuel/Sq Ft Grate/Hr...... lb
45. Actual Water Evaporated per Hour .... lb
46. Correction for Uncovered Boiler per Hour........ lb
47. Total Equivalent Evaporation per Hour .... lb

OUTPUT

48. Output in Btu per Hour........ Btu
49. Output in Sq Ft of Steam Radiation........ sq ft
50. Actual Evaporation per Pound of Fuel, as Fired........ lb
51. Equivalent Evaporation per Pound of Fuel, as Fired........ lb

HEAT BALANCE

52. Heat Transferred by Water (efficiency of boiler, furnace and grate). per cent
53. Heat Carried Away by Steam in Flue Gases........ per cent
54. Heat Carried Away by Dry Flue Gases .... per cent
55. Heat Lost by Not Burning Carbon Monoxide........ per cent
56. Heat Lost by Not Burning Combustible in Ash........ per cent
57. Heat Unaccounted for .... per cent
58. Efficiency of Boiler and Furnace........ per cent

Additional Items, for Use Only with New-Fire Method of Starting

FUEL USED

59. Weight of Wood for Kindling ....lb
60. Fuel Equivalent of Wood ....lb
61. Total Fuel Fired during Test, Exclusive of Wood....................lb
62. Total Equivalent Fuel Charged during Test..........................lb
63. Weight of Residual Fire ....lb
64. Heating Value per Pound of Residual Fire.........................Btu
65. Fuel Equivalent of Residual Fire ....lb
66. Equivalent Fuel Used during Test ....lb

COMPUTATIONS FOR TEST OF STEAM HEATING BOILER FOR TABLE 1

EXPLANATION OF SYMBOLS

Wherever CO_2, O_2, CO and N_2 are used they are the percentages by volume of these constituents in the gases of combustion.

A_r = the allowed projected area of uncovered metal steam-heated surface

H = total heat (in Btu per lb) of saturated steam at the boiler outlet pressure

h = total heat (in Btu per lb) in feedwater at boiler inlet

L = latent heat (in Btu per lb) of steam at the boiler outlet pressure

t_s = temperature of steam and boiler outlet pressure

t_w = temperature of feedwater and boiler inlet

FORMULAE AND NOTES

Item 19 = When tests are made by the new-fire method, the value given by *Item* 66 is to be used for *Item* 19

Item 20 = Weight of fuel fired during test (minus 1/3 charged in first firing in new-fire method) divided by number of firings

Item 21 = Total time of test divided by number of firings

Item 22 = Total time of test divided by number of attention periods.

$$Item\ 25 = \frac{Item\ 13}{100} - \frac{Item\ 23 \times Item\ 24}{100 \times Item\ 19}$$

$$Item\ 35 = \frac{700 + 4\ CO_2 + O_2}{3\ (CO_2 + CO)} \times Item\ 25$$

Item 38 = (When condenser is used) weight of water condensed + *Item* 39

$$Item\ 40 = \frac{Item\ 39 \times 100}{Item\ 38}$$

$$Item\ 41 = Item\ 38 - Item\ 39$$

$$Item\ 42 = \frac{L + t_s - t_w}{971.7} \text{ (for saturated or wet steam)}$$

$$Item\ 43 = \frac{Item\ 19}{Item\ 4 \times Item\ 5}$$

$$Item\ 44 = \frac{Item\ 43 \times Item\ 18}{12{,}500}$$

$$Item\ 45 = \frac{Item\ 41}{Item\ 5}$$

$$Item\ 46 = \frac{1.8}{H - h}(T_s - Item\ 34) \times A_r$$

$$Item\ 47 = (Item\ 45 \times Item\ 42) + Item\ 46$$

$$Item\ 48 = Item\ 47 \times 971.7$$

$$Item\ 49 = \frac{Item\ 48}{240}$$

$$Item\ 50 = \frac{Item\ 41}{Item\ 19}$$

$$Item\ 51 = \frac{Item\ 47 \times Item\ 5}{Item\ 19}$$

$$Item\ 52 = \frac{Item\ 51 \times 971.7 \times 100}{Item\ 18}$$

$$Item\ 53 = \frac{9 \times (1089 + 0.46 \times Item\ 33 - Item\ 34) \times Item\ 12}{Item\ 18}$$

$$Item\ 54 = \frac{Item\ 35 \times 0.24\ (Item\ 33 - Item\ 34) \times 100}{Item\ 18}$$

$$Item\ 55 = \frac{\frac{CO}{CO_2 + CO} \times Item\ 25 \times 10{,}150 \times 100}{Item\ 18}$$

$$Item\ 56 = \frac{Item\ 23 \times Item\ 24 \times 14{,}540}{Item\ 19 \times Item\ 18}$$

$$Item\ 57 = 100 - (Item\ 52 + Item\ 53 + Item\ 54 + Item\ 55 + \text{Item}\ 56)$$

$$Item\ 58 = \frac{Item\ 52 \times 100}{100 - Item\ 56}$$

$$Item\ 60 = \frac{Item\ 59 \times 5000}{Item\ 18}$$

$$Item\ 62 = Item\ 60 + Item\ 61$$

$$Item\ 65 = \frac{Item\ 63 \times Item\ 64}{Item\ 18}$$

Item 66 = *Item* 62 — *Item* 65. This value to be used for *Item* 19

CODE NO. 2

Short Test Form

TABLE 2. DATA AND RESULTS OF TEST—SOLID FUELS

GENERAL INFORMATION

1. Number of Test ....
2. Date of Test ....
3. Maker and Catalog Designation of Boiler.............................
4. Grate Surface ....sq ft
5. Duration of Test ....hour

FUEL

6. Kind ....
7. Size ....

Proximate Analysis, as fired:

8. Moisture ....per cent
9. Volatile Matter ....per cent
10. Fixed Carbon ....per cent
11. Ash ....per cent
12. Heating Value per pound as Fired ....Btu
13. Total Fuel Fired during Test ....lb

14. Average Fuel per Firing ....lb
15. Average Interval between Firings ....hour
16. Average Interval between Attention to Fire ....hour

ASH AND REFUSE

17. Total Ash and Refuse Removed during Test ....lb
18. Combustible in Ash and Refuse ....per cent

DRAFT

19. In Ashpit ....in. of water
20. In Furnace ....in. of water
21. In Stack ....in. of water

FLUE GASES

22. Carbon Dioxide in Dry Flue Gases ....per cent
23. Temperature of Flue Gases Leaving Boiler............................F

STEAM AND FEED WATER

24. Steam Pressure by Gage, Boiler..............................lb per sq in.
25. Temperature of Feed Water Entering Boiler...........................F
26. Total Weight of Water Fed to Boiler.................................lb
27. Total Weight of Water from Separator................................lb
28. Percentage of Moisture in Steam ....per cent
29. Total Water Evaporated ....lb
30. Factor of Evaporation ....

HOURLY RATES

31. Fuel Burned per Sq Ft of Grate per Hour, as Fired...................lb
32. Equivalent Rate of Combustion of 12,500 Btu Fuel/Sq Ft Grate/Hr......lb
33. Actual Water Evaporated per Hour ....lb
34. Correction for Uncovered Boiler per Hour............................lb
35. Total Equivalent Evaporation per Hour ....lb

OUTPUT

36. Output in Btu per Hour ....Btu
37. Output in Sq Ft of Steam Radiation ....sq ft
38. Actual Evaporation per Pound of Fuel, as Fired......................lb
39. Equivalent Evaporation per Pound of Fuel, as Fired..................lb

EFFICIENCY

40. Heat Transferred to Water (efficiency of boiler, furnace and grate)..per cent
41. Heat Loss by Not Burning Combustible in Ash..................per cent
42. Efficiency of Boiler and Furnace.............................per cent

Additional Items, for Use Only with New-Fire Method of Starting

FUEL USED

43. Weight of Wood Used for Kindling ....lb
44. Fuel Equivalent of Wood ....lb
45. Total Fuel Fired during Test, Exclusive of Wood.....................lb
46. Total Equivalent Fuel Charged during Test...........................lb
47. Weight of Residual Fire ....lb
48. Weight of Combustible Burned ....lb
49. Equivalent Fuel (as fired basis) Used during Test...................lb

COMPUTATIONS FOR SHORT FORM REPORT OF TESTS ON STEAM HEATING BOILERS

EXPLANATION OF SYMBOLS

A_r = the allowed projected area of uncovered metal steam-heated surface
H = total heat (in Btu per lb) of saturated steam at the boiler outlet pressure
h = total heat (in Btu per lb) in feedwater at boiler inlet

L = latent heat (in Btu per lb) of steam at the boiler outlet pressure
t_a = room temperature
t_s = temperature of steam and boiler outlet pressure
t_w = temperature of feedwater and boiler inlet

GENERAL

Item 13 = When tests are made by the new-fire method the value given by *Item* 49 is to be used for *Item* 13
Item 14 = Weight of fuel fired during test (minus 1/3 charged in first firing in new-fire method) divided by number of firings
Item 15 = Total time of test divided by number of firings
Item 16 = Total time of test divided by number of attention periods

$$Item\ 28 = \frac{Item\ 27 \times 100}{Item\ 26}$$

$$Item\ 29 = Item\ 26 - Item\ 27$$

$$Item\ 30 = \frac{L + t_s - t_w}{971.7} \text{ (for saturated or wet steam)}$$

$$Item\ 31 = \frac{Item\ 13}{Item\ 4 \times Item\ 5}$$

$$Item\ 32 = \frac{Item\ 31 \times Item\ 12}{12{,}500}$$

$$Item\ 33 = \frac{Item\ 29}{Item\ 5}$$

$$Item\ 34 = \frac{1.8}{H - h}\ (t_s - t_a) \times A_s,$$

$$Item\ 35 = (Item\ 33 \times Item\ 30) + Item\ 34$$

$$Item\ 36 = Item\ 35 \times 971.7$$

$$Item\ 37 = \frac{Item\ 36}{240}$$

$$Item\ 38 = \frac{Item\ 33}{Item\ 13}$$

$$Item\ 39 = \frac{Item\ 35}{Item\ 13}$$

$$Item\ 40 = \frac{Item\ 39 \times 971.7 \times 100}{Item\ 12}$$

$$Item\ 41 = \frac{Item\ 17 \times Item\ 18 \times 14{,}540}{Item\ 13 \times Item\ 12}$$

$$Item\ 42 = \frac{Item\ 40 \times 100}{100 - Item\ 41}$$

$$Item\ 44 = \frac{Item\ 43 \times 5000}{Item\ 12}$$

$$Item\ 46 = Item\ 44 + Item\ 45$$

$$Item\ 48 = Item\ 46 - Item\ 47 - Item\ 17$$

$$Item\ 49 = \frac{Item\ 48 \times 100}{100 - Item\ 11}$$. This value to be used for *Item* 13

Log sheets for recording various observations in detail are same as 1923 Edition of Code

CODE NO. 3

A.S.H.&V.E. PERFORMANCE TEST CODE FOR STEAM HEATING SOLID FUEL BOILERS[1]

EDITION OF JUNE, 1929

1. *The object* of this code is to specify the tests to be conducted and to provide a standard method for conducting and reporting tests to determine the efficiencies and performance of the boiler.

2. *Chimneys.*—Tests may be conducted with either natural draft or induced draft. The boiler shall preferably be attached to a chimney of dimensions nearest to those cataloged by the manufacturer. If a proper sized chimney is not available and therefore induced draft or a larger chimney is used, means must be provided for reducing the draft so that the draft at no time exceeds that which would have been obtained with chimney as specified by the manufacturer.

3. *Boiler Set-Up.*—The boiler tested shall be a standard stock boiler. It shall be assembled in accordance with the manufacturer's directions, care being taken that all openings which would normally be closed (when manufacturer's directions are followed) in practice are closed with the proper type of cement or putty.

4. *If the boiler* is provided with an insulated jacket, this jacket shall be in place during tests. If boiler is uninsulated, all parts which would normally be covered in practice shall be covered with a coating of asbestos cement 1½ in. thick or equivalent insulation, or the boiler may be tested without covering and correction applied as explained in Par. 40 (Item *n*) under Computations. Parts of boiler which are not insulated in practice shall not be insulated during tests and shall not be considered as external boiler surface.

ARRANGEMENT OF APPARATUS

5. *The general arrangement* of apparatus shall be as shown in Fig. 1. Minimum apparatus for conducting tests shall be the following:

(*A*)—SCALES for weighing coal and residue

(*B*)—CALIBRATED weighing tank or other means for determining water evaporated

(*C*)—THERMOMETERS for determining feed water and steam temperature

(*D*)—MERCURY manometer for determining steam pressure

(*E*)—DRAFT gages for determining ashpit, fire box, and smoke outlet draft

(*F*)—PYROMETER for determining temperature of gases leaving boiler

(*G*)—RECORDING or indicating CO_2 and CO apparatus and means for checking recording apparatus

(*H*)—SCALE for weighing condensation obtained from separator

(*I*)—BAROMETER or barograph for determining atmospheric pressure during test.

6. *Location of Instruments.*—The feed water thermometer shall be inserted in a thermometer well so located as to read the true temperature of the water entering the boiler. The steam temperature thermometer shall be inserted directly into the steam in the flow outlet.

7. *The mercury manometer* shall be connected with the steam space of the boiler.

8. *A draft gage* shall be connected to an open end tube set at right angles to the flow of gas in each of the following: ashpit, fire box, and smoke outlet collar. To prevent influence upon gas samples, draft reading, or gas temperature reading, choke dampers between flues and smoke outlet collar shall be left in wide open position throughout all tests. Cold air checks between smoke outlet collar and flues shall be thoroughly sealed during tests. Air leakage around draft tubes, thermocouples and gas sampling tubes shall be prevented by using asbestos rope or other suitable means packed around the tubes.

9. *The temperature* of gases leaving the boiler shall be taken by means of an

[1] Code prepared by F. B. Howell, *Chairman*, R. V. Frost, T. E. Langvoit, Carl H. Flink and ex-officios, Edwin W. Smith and Frederick W. Herendeen; adopted as Steam Heating Boiler Testing Code by the *National Boiler and Radiator Manufacturers' Association*.

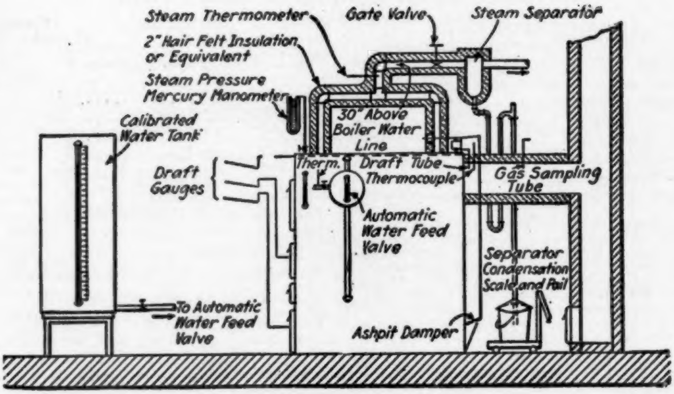


FIG. 1—ARRANGEMENT OF TEST APPARATUS

exposed thermocouple located as shown on Fig. 2. The smoke hood and smoke pipe shall be covered with 2 in. of asbestos cement to a distance 1 ft beyond the thermocouple.

10. *The gas* sampling tube shall be a single open end tube in position corresponding to gas temperature thermocouple and shall be placed in the smoke pipe within 1 ft of the gas temperature thermocouple as shown on Fig. 2.

11. *Accuracy of Instruments.*—All instruments shall be calibrated to insure accuracy.

12. *Fuels to be Used.*—For anthracite boilers the fuel used shall be the commercial size, best suited to the boiler, as specified by the manufacturer. For soft coal boilers the coal used shall be a free burning 3 x 2 in. lump coal unless the boiler performs more satisfactorily with caking coal. If caking coal is used it shall be of approximately 3 x 2 in. lump size. For coke boilers the coke used shall be by-product or gas coke of commercial size, best suited to the boiler.

13. *If the boiler* is designed for a special solid fuel, tests shall be conducted with this fuel. In all cases the characteristics, kind, and size of fuel used in making tests shall be recorded on the test form. Sizes of fuel used shall be according to classification of *American Society of Mechanical Engineers,* as given in Table 1.

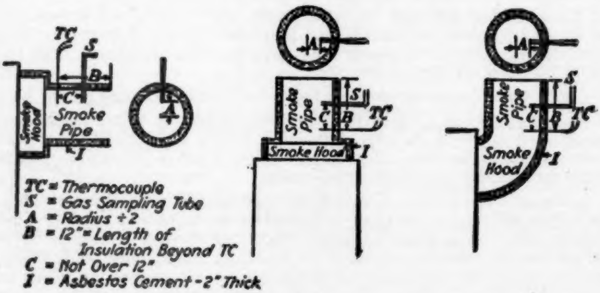


FIG. 2—LOCATION OF GAS TEMPERATURE THERMOCOUPLE AND GAS SAMPLING TUBE

TABLE 1. NAMES AND SIZES OF VARIOUS FUELS

DOMESTIC HARD COALS

Name and Size	Screen Through	Screen Over
Pea	¾	½
Chestnut or Nut	1¼	¾
Stove or Range	1¾	1¼
Egg—in the East	2½	1¾
Large Egg—Chicago	4	2¾
Small Egg—Chicago	2¾	2
Broken or Grate	4	2½

DOMESTIC BY-PRODUCT COKE

Name and Size	Screen Through	Screen Over
Egg	3	2½
Large Stove	2½	2
Small Stove	2	1½
Nut	1½	¾
Pea	¾	½

BITUMINOUS COAL

Name and Size	Screen Through	Screen Over
Duff	⅛	...
No. 3 Nut	1¼	¾
No. 2 Nut	2	1¼
No. 1 Domestic Nut	3	1½ or 2
No. 4 Washed	¾	¼
No. 3 Washed Chestnut	1¼	¾
No. 2 Washed Stove	2	1¼
No. 1 Washed Egg	3	2
No. 3 Roller Screened Nut	1½	1
No. 2 Roller Screened Nut	2	1½
No. 1 Roller Screened Nut	3½	2
Egg	6	3
Lump or Block	...	6

Run-of-Mine: fine and large lumps.

Screenings: usually smallest sizes.

Pocahontas Smokeless: Generally sized as Nut, Egg, Lump, and Mine Run.

Cannel Coal:
For fireplaces—Hand Picked Lump.
For stoves—Egg.

TESTS TO BE CONDUCTED

14. *Five tests* shall be conducted at 2 lb steam gage pressure—one shall be at drive output and the other four at various output rates to show the entire operating range of the boiler. The manufacturer shall determine and designate the output rates and the operating range of the boiler. No test data or boiler information shall be assumed for an output beyond the drive test. The drive test shall be considered the boiler's maximum output. No tests shall be reported in which the water collected from the separator is greater than 2 per cent of the water fed to the boiler in excess of the condensation resulting from the normal heat loss of the separator and piping. Correction for the water collected must be applied to the water evaporated in order to include in the output only the heat actually absorbed to heat the water from feed water temperature to steam temperature.

15. *Outlet Piping.*—The number and size of outlets shall be as cataloged by the manufacturer. Outlets shall be full size to a height 30 in. above the boiler's normal water line. An effective separator shall be placed in the steam outlet as near the boiler as practicable. The separator shall be open to the atmosphere through a water seal permitting any water separated from the steam to drain immediately into a container for weighing. Re-evaporation of water drained from separator shall be prevented by use of a suitable cover on the container.

16. *A gate valve,* full size of flow piping, shall be placed between boiler and separator for the purpose of maintaining a constant steam pressure in the boiler at various test output rates. All piping up to and including separator shall be insulated with 2 in. of hair felt, or equivalent.

17. *Outlet piping* shall pitch away from the risers and away from the separator.

18. *For all boilers* up to 3,000 sq ft of steam radiation, water shall be supplied by an automatic feed valve of the slow feed type, which will maintain a constant water level at the normal water line. For boilers above 3,000 sq ft output, feed water may be regulated by hand by means of a gate valve.

19. *Chimney Connection.*—Connection from smoke hood outlet to chimney shall be of size cataloged by manufacturer and shall be air tight and as short and direct as possible, preference being given to long radius and 45 deg instead of 90 deg bends. The bend at entrance to chimney shall be carefully made, care being taken that the pipe does not project beyond inside of chimney and that no leakage of air occurs.

20. *Cleaning Boilers.*—The water and steam surfaces of the boiler shall be thoroughly cleaned by vigorous boiling with a solution of sal-soda or lye, the scum form-

SHEET NO.________OF________SHEETS

BOILER ROOM LOG SHEET—NEW FIRE METHOD

BOILER
- Name and Number
- Description
- Manufactured by
- Object of Test

DESIRED OUTPUT
- Sq. Ft.
- % of Rating
- Lbs. of Water

Date of Test
Serial No. ________ Test No.
Fuel { Kind / Size
Chimney: Size (dia.) Height (ft.)
Induced Draft

FUEL ANALYSIS

Proximate	%	Ultimate	%
Volatile Matter		Carbon	
Fixed Carbon		Hydrogen	
Ash		Nitrogen	
Sulphur		Oxygen	
		Sulphur	

Calorific Value B.t.u. per lb.
Steam Outlets Used: No. Size (ins.)
Location
Grate Area* (sq. ft.) Heating Surface (sq. ft.)
*Fire box at Grate Level External Surface (sq. ft.)

		POUNDS
Fuel Capacity at	ins. fuel depth	
Fuel Available	% of above capacity)	
a. Wood used in starting fire		
b. Fuel equivalent of wood	(a)×0.4	
c. Fuel fired during test		
d. Total fuel as fired	(b)+(c)	
e. Moisture in fuel %	lbs.	
f. Total dry fuel fired	(d)−(e)	
g. Total residue and cleanings		
h. Combustible burned	(f)−(g)	
i. Net dry fuel burned	(h)+(l)	
j. Dry fuel burned per hour	(i)÷hrs.	
k. Dry fuel burned/hour/sq. ft. grate	(j)÷G.A.	
l. Combustible per pound of fuel from Analysis		
Cleanings from Ashpit: Total		
Ash (lb.) Clinker (lb.)		
Unburned (lb.) Slate (lb.)		
Clinker from Firebox		

	Hr. of Test	TIME Hrs.	TIME Min.	Reading on Tank Gage	WATER (LBS.) Reading on Tank Gauge	Evap. between readings	Evap. per hour	Total Evap. from start of Test	DRAFTS (ins. H_2O) In smoke Outlet	In Fire-box	In ash pit						Barom. Pressure Ins. Hg.	Steam Pressure Lbs. per Sq. In.	TEMPERATURES °F Feed Water	Steam	In Smoke Outlet	Boiler Room	Outside Air
Total																							
Average																							

		POUNDS
m. Actual evaporation x factor of evaporation		
n. Corrections | Heat loss from uncovered boiler		
o. | Heat loss from uncovered piping		
p. Total Equivalent Evaporation	(m)+(n)+(o)	
q. Equivalent Evaporation per hr.	(p)÷hrs.	
r. Output in Sq. Ft. of Radiation	(q)×970.4÷240	
s.		

t. Boiler Eff. %
u. B & G Eff. %

Other side for additional remarks, fuel weights, residue and gas anal.

FORM A

SHEET NO ______ OF ______ SHEETS

BOILER ROOM LOG SHEET—CONTINUOUS FIRE METHOD

BOILER — Name and Number; Description; Manufactured by; Object of Test

DESIRED OUTPUT — Sq. Ft.; % of Rating; Lbs. of Water

Date of Test
Serial No. Test No.
Fuel { Kind; Size
Chimney: Size (dia.) Height (ft.)
Induced Draft

FUEL ANALYSIS

Proximate	%	Ultimate	%
Volatile Matter		Carbon	
Fixed Carbon		Hydrogen	
Ash		Nitrogen	
Sulphur		Oxygen	
		Sulphur	

Calorific Value B.t.u. per lb.
Steam Outlets Used: No. Size (ins.)
Location
Grate Area* (sq. ft.) Heating Surface (sq. ft.)
*Fire box at Grate Level External Surface (sq. ft.)

		POUNDS
Fuel Depth—Beginning and end of test	inches	
a. Total fuel as fired		
b. Moisture in fuel %	lbs.	
c. Total dry fuel fired	(a)−(b)	
d. Cleanings from ashpit		
e. Clinker from fire box		
f. Total cleanings	(d)+(e)	
g. Combustible in total cleanings %	lbs.	
h. Fuel equivalent of combustible in cleanings	(g)×14600 / cal. value	
i. Net dry fuel burned	(c)−(h)	
j. Dry fuel burned per hour	(i)÷hrs.	
k. Dry fuel burned per hour per sq. ft. grate	(j)÷G.A.	

	Nr. of Test	TIME Hrs.	TIME Min.	Reading on Tank Gauge	WATER (LBS.) Reading on Tank Gauge	WATER (LBS.) Drop between readings	WATER (LBS.) Drop per hour	WATER (LBS.) Total Drop from start of Test	DRAFTS (ins. H_2O) In smoke Outlet	DRAFTS (ins. H_2O) In Fire-box	DRAFTS (ins. H_2O) In ash pit						Steam Pressure Ins. Hg.	Steam Pressure Lbs. per Sq. In.	TEMPERATURES °F Feed Water	TEMPERATURES °F Steam	TEMPERATURES °F In Smoke Outlet	TEMPERATURES °F Boiler Room	TEMPERATURES °F Outside Air
Total																							
Average																							

	POUNDS
m. Actual evaporation x factor of evaporation	
n. Corrections | Heat loss from uncovered boiler	
o | Heat loss from uncovered piping	

t. Boiler Eff. %

u. B & G Eff. %

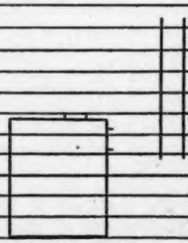
		POUNDS
p. Total Equivalent Evaporation	(m)+(n)+(o)	
q. Equivalent Evaporation per hr.	(p)÷hrs.	
r. Output in Sq. Ft. of Radiation	(q)×970.4÷240	
s.		

Other side for additional remarks, fuel weights, residue and gas anal

FORM B

WOOD				RESIDUE			
TIME FIRED	GROSS	TARE	NET	TIME WEIGHED	GROSS	TARE	NET
COAL							

GAS ANALYSIS				TEST NOTES
TIME	CO_2	O	CO	



Sketch of Connection to Chimney

READINGS AND WEIGHTS, RECORDED BY

FORM C

ing at the surface of the water being blown out through a surface blow out, or the boiler outlets, with fresh water gradually fed to the boiler, until the water in the gage glass appears clean. The boiler shall then be drained and refilled with clean water.

21. *Duration of Test.*—Each test shall be conducted at the discretion of the engineer until there has been burned in pounds per square foot of grate not less than 250 in the continuous fire method or 150 in the new fire method, or until the error in fuel and water determination shall not exceed 2 per cent.

22. *Calorific Tests and Analysis of Fuels and Refuse.*—The quality of the fuel and of the ash and refuse shall be determined by calorific tests and analysis of samples. Directions for obtaining samples of the fuel and making these tests and analyses shall be the recommendations in the *A. S. M. E. Test Code for Solid Fuels*—April 1927.

23. *A representative sample* of the fuel shall be taken during the test and shall be kept in an air tight container pending determination of moisture.

Starting and Stopping of Test

24. *New Fire Method.*—New fire method of starting and stopping test may be used on anthracite or coke boilers. A preliminary fire using wood or coal shall be made and the boiler shall be operated until the water in the boiler has reached steam temperature and the boiler insulation normal operating temperature. During this preliminary operation the water in the gage glass shall be kept at the normal boiler water line. The fire shall then be quickly removed from the boiler, the ashpit cleaned, the flues thoroughly cleaned, the quantity of water in tank noted, and the test started by placing a charge of kindling wood on the grate (9 lb for each sq ft of grate) igniting same and firing 33½ per cent of the *Fuel Charge* (*Available Fuel*). A second amount of fuel equal to the *Fuel Charge* (*Available Fuel*) shall be fired within the first half hour of test. No firing or attention to fuel shall be permitted after this first half hour period until the time of adding the next fuel charge.

25. *When test* has been conducted for such a period as specified under *Duration of Tests* (Par. 21), and the fuel has burned down to that point where the output can no longer be maintained by operation of the dampers, the test shall be ended as follows:

(*A*)—The grates shall be shaken to remove ash from the fire as if another charge of fuel were to be added

(*B*)—The cleanings shall be quickly removed from the ash pit and weighed

(*C*)—The residue within the the fire box shall be quickly quenched with just sufficient water to put out the fire. (The quantity of water used shall be recorded.)

(*D*)—The entire residue of the fire box shall be removed and weighed, and shall be kept separate from the cleanings. The water level at the end of the test shall be the same as at the beginning of test.

26. *Continuous Fire Method.*—The continuous fire method may be used for all boiler tests. The boiler shall be operated under test conditions for at least one firing period and not less than one hour before the starting of test. A preliminary fire shall be made and the boiler operated until the water has reached steam temperature, and the boiler insulation normal operating temperature. The flues shall then be thoroughly cleaned. The fire shall then be burned low, thoroughly cleaned, and the remaining live fuel spread evenly over the grate as the foundation for the first test fuel charge. If the manufacturer provides firing directions these directions shall be followed.

27. *The thickness* of the fuel bed shall be quickly measured and the condition of the fuel bed noted. The height of water in boiler and feed tank shall be recorded. Tests shall start at the time of making these observations. The first weighed charge of fuel (see Par. 29) shall then be fired. If firing directions are specified by manufacturer, same shall be followed. The ashpit shall be thoroughly cleaned immediately and the test allowed to proceed.

28. *At the end* of the test fire shall be burned low and cleaned so as to leave the fuel bed in the same condition and thickness as at the start of the test. When this condition is reached the water level in the boiler shall be at the same height as at the start and the test shall be ended at this time. The contents of the ashpit shall be quickly removed and weighed.

HEATING BOILER TEST REPORT—SOLID FUEL BURNING

Boiler			Connection to Chimney—Length		Fuel Capacity (Catalog) Lbs.		B. T. U.					
Rating Sq. Ft.			" " " —No. of Bends		Fuel Available " Lbs.		Moisture					
Grate Area Sq. Ft.			" " " —Size		Fuel Depth " Ins.		Volatile Matter					
Heating Surface Sq. Ft.							Fixed Carbon					
							Ash					
							Sulphur					

Date of Test															
Serial Number															
Test Number															
1 OUTPUT	Sq. Ft. Radiation	Sq. Ft.													
2	Percent of Rating	%													
3 EFFICIENCY	Boiler	%													
4	Boiler and Grate	%													
5															
6 STACK GAS TEMPERATURE		F°													
7	Outside Air Temperature	F°													
8 LOSSES	Grate	%													
9	In Dry Gases Leaving Boiler	%													
10	Incomplete Combustion	%													
11	Radiation from Covered Boiler	%													
12 FUEL	Kind														
13	Size														
14	Calorific Value	B. T. U.													
15	Burned per hour	Lbs.													
16	Burned per sq. ft. grate per hour	Lbs.													
17	Average Interval Between Firings*	Hrs.													
18	Average Interval Between Attention	Hrs.													
19	Clinker	%													
20 GAS ANALYSIS	CO_2	%													
21	O	%													
22	CO	%													
23 DRAFT	Natural or Induced														
24 DRAFT	In Smoke Outlet	Inches H_2O													
25	Over Fire	" "													
26	In Ash Pit	" "													
27 CHIMNEY	Size	Inches													
28	Height	Ft.													
29 Duration of Test		Hrs.													

*Based on coal of 12500 B. t. u. per pound.

BOILER MFR.

FORM D

29. *The Fuel Charge* for ALL tests shall be:

(*A*)—The amount specified by the manufacturer, or

(*B*)—The amount determined by the engineer in charge of test as the maximum amount of fuel consistent with proper boiler performance.

30. *The Fuel Charge* (*Available Fuel*) used in computations to determine the *Average Interval between Firings* which is recorded on Form D, Item 17, shall be obtained as follows:

(*A*)—NEW FIRE METHOD: Divide the total quantity of fuel fired during test (but not including the wood or coal used as a preliminary charge) by the number of firings during test.

(*B*)—CONTINUOUS FIRE METHOD: Divide the total quantity of fuel fired during test by the number of firings during test.

31. *Attention to fire* shall be considered as any one or more of the following operations and in all tests burning anthracite or coke of nut or larger size there shall be no shaking of grates or other attention to fire except at time of recharging boiler.

(*A*)—FIRING

(*B*)—STOKING

(*C*)—LEVELING

(*D*)—SHAKING GRATE

32. *The average interval* between the end of any attention period and the beginning of the next shall be stated in the report of test.

33. *All dampers* controlling the primary and secondary air may be adjusted as required to produce most efficient combustion at any time during test.

34. *The following readings* shall be recorded at regular intervals not greater than 15 min throughout test:

(*A*)—WATER evaporated

(*B*)—TEMPERATURE of feed water

(*C*)—STEAM temperature

(*D*)—STEAM pressure

(*E*)—DRAFTS in ash pit, fire box and smoke outlet

(*F*)—TEMPERATURE of gases leaving boiler

(*G*)—WEIGHT of water from separator

35. *Following readings* shall be recorded every hour:

(*A*)—BOILER room temperature

(*B*)—OUTSIDE temperature

(*C*)—BAROMETRIC pressure

36. *In all tests* using bituminous coal, any smoke observations shall be recorded every 15 seconds according to Ringelmann Chart numbers throughout two complete firing periods.

TEST FORMS

37. *Results of tests* shall be recorded on test forms as follows:

(*A*)—BOILER ROOM LOG SHEET—Form A (for new fire method) or Form B (for continuous fire method) shall be used for recording all readings during test.

(*B*)—THE BACK OF THE BOILER ROOM LOG SHEET—Form C shall be used for recording weights of fuel, residue, gas analysis (if Orsat Apparatus only is used), or for recording check of recording gas analysis apparatus with Orsat Apparatus.

38. *Results* of all tests on one boiler shall be reported on Form D.

39. *The following results* referred to boiler output in square feet of steam radiation, taken from Form D shall be shown by curves on Form E:

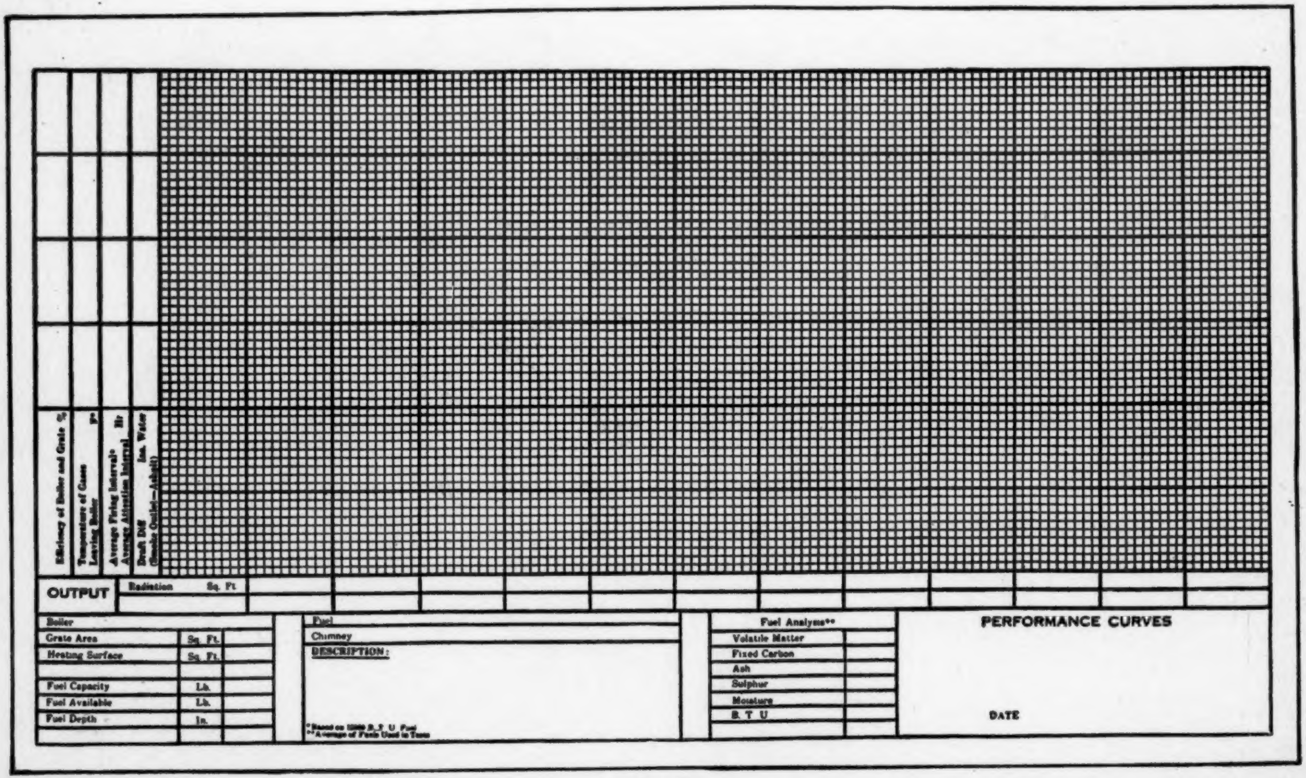
(*A*)—BOILER and grate efficiency

(*B*)—TEMPERATURE of gases leaving boiler

(*C*)—AVERAGE firing interval

(*D*)—AVERAGE attention interval

(*E*)—DRAFT difference (smoke outlet minus ashpit).


Efficiency of Boiler and Grate %
Temperature of Gases Leaving Boiler F°
Average Firing Interval* Average Attention Interval Hr
Draft Diff (Smoke Outlet—Ashpit) Ins. Water
OUTPUT
Radiation Sq. Ft.
Boiler
Grate Area Sq. Ft.
Heating Surface Sq. Ft.
Fuel Capacity Lb.
Fuel Available Lb.
Fuel Depth In.
Fuel
Chimney
DESCRIPTION:
Fuel Analysis**
Volatile Matter
Fixed Carbon
Ash
Sulphur
Moisture
B. T. U
PERFORMANCE CURVES
DATE

FORM E

COMPUTATIONS

40. *Form A.*

ITEM (b) = The factor 0.4 is the weight of coal equivalent in calorific value to 1 lb of wood.

ITEM (n) = (External Boiler Surface) × (1.8) × $(T_s - T_r)$ × (hrs) × 1/971.7.
1.8 = (Coef. of uncovered boiler surface—coef. of covered boiler surface) = Btu/sq ft hr degrees Fahrenheit diff. of surface and room.
T_s = Surface temperature = steam temperature.
T_r = Room temperature.
hrs = Duration of test.

ITEM (o) = Surface × 2.4 × $(T_s - T_r)$ × (hrs) × (1/971.7).
(Surface) = External surface of pipe in sq ft
2.4 = Average coefficient for uncovered pipe.
T_s = Steam temperature.
T_r = Room temperature.
hrs = Duration of test.

$$\text{ITEM (t)} = \frac{\text{ITEM }(p) \times 971.7}{\text{ITEM }(i) \times \text{(Calorific value)}} \times 100$$

$$\text{ITEM }(u) = \frac{\text{ITEM }(p) \times 971.7}{[\text{ITEM }(i) + \text{Fuel through grate}] \times \text{(Calorific value)}} \times 100$$

41. *Form B.*

ITEM (h) = 14,600 = Calorific value of 1 lb carbon.
It is assumed that the combustible in ash is carbon.

$$\text{ITEM }(u) = \frac{\text{ITEM }(p) \times 971.7 \times 100}{\text{ITEM }(c) \times \text{Calorific value}}$$

ITEM (n), (o) and (t) same as shown under Form A.

42. *Form D.*

$$\text{ITEM 8} = \frac{\text{Fuel through grate}}{\text{Dry fuel burned} + \text{fuel through grate}} \times 100 \text{ (New Fire Method)}$$

or

$$\frac{\text{Fuel through grate}}{\text{Dry fuel fired}} \times 100 \text{ (Continuous Fire Method)}$$

$$\text{ITEM 9} = \frac{11\,CO_2 + 8\,O_2 + 7\,(CO + N_2)}{3\,(CO_2 + CO)} \times \frac{C \times 0.24 \times (T_f - T_r)}{\text{Calorific value}}$$

CO_2, O_2, CO and N are in o/o by volume obtained and
C is taken as per cent carbon by weight per pound of fuel.
T_f = Temperature of gases leaving boiler.
T_r = Room temperature.

$$\text{ITEM 10} = \frac{CO}{CO_2 + CO} \times C \times \frac{10{,}160}{\text{Cal. value}} =$$

CO_2, CO and C same as for ITEM 9.

$$\text{ITEM 11} = \frac{\text{Surface of boiler} \times \text{Coef. of covered boiler surface} \times (T_s - T_r) \times 100}{\text{Fuel burned per hour} \times \text{Calorific value}}$$

$$\text{ITEM 17} = \frac{\text{Fuel charge} \times \text{(boiler and grate efficiency)} \times 12{,}500}{\text{Output} \times 240}$$

$$\text{ITEM 18} = \frac{\text{Hours}}{\text{Times given attention}}$$

$$\text{ITEM 19} = \frac{\text{Weight of clinker removed}}{\text{Dry fuel burned} + \text{Fuel through grate}} \times 100 \text{ (New Fire Method)}$$

or

$$\frac{\text{Weight of clinker removed}}{\text{Dry fuel fired}} \times 100 \text{ (Continuous Fire Method)}$$

DEFINITION OF TERMS USED IN THE CODE

43. *Boiler Efficiency* is the percentage of the heat in the net dry fuel burned delivered at the boiler nozzle when the boiler is normally insulated.

44. *Boiler and Grate Efficiency* (*Overall Efficiency*) is the percentage of the heat in the net dry fuel burned plus the heat in the fuel fallen through the grate delivered at the boiler nozzle when the boiler is normally insulated.

45. *Fuel Charge* (*Available Fuel*) is the maximum amount of fuel that can actually be fired at one time and can be burned with proper boiler performance between firing intervals. It must be demonstrated during tests that the quantity cataloged by the manufacturer as *Fuel Charge* (*Available Fuel*) can actually be fired at one time upon a fuel bed that is sufficiently deep to properly ignite the *Fuel Charge*. The following values for weight of fuel per cu ft shall be used for catalogue purposes:

Anthracite	52.5
Bituminous	45.0
Coke	28.0

46. *Fuel Capacity* is the weight of fuel that must be left in the boiler to kindle a fuel charge plus the weight of fuel which can be fired at one time. When specifying fuel capacity, it is necessary to specify the kind of fuel. Fuel capacity is not the total weight of fuel that can be shoveled into the fuel and combustion space because there must be sufficient furnace volume in addition to that occupied by the fuel to permit the escape of the gases leaving the fuel.

47. *Calorific Value* is the number of Btu per pound of dry fuel as determined in a bomb calorimeter.

48. *Clinker.*—(ITEM 19, Form D) is the weight of clinkers which must be removed through fire or clinker doors. It does not include any clinker that drops into ashpit either during normal operation or when grates are shaken.

49. *Residue.*—(ITEM *g*, Form A) is the weight of the contents of the firebox when ending a test in the new fire method.

50. *Cleanings.*—(ITEM *g*, Form A) is the weight of material that drops through or is shaken through the grate into the ashpit.

51. *Combustible per Pound of Fuel from Analysis.*—(ITEM *l*, Form A) equals one minus the weight of ash obtained from the analysis of the dry fuel.

52. *External Surface* (Form A and Form B) is the measured surface of the boiler which is normally insulated in an installation.

53. *Radiation from Covered Boiler* (ITEM *ll*, Form D) is the heat lost through insulated surfaces of the boiler. It is not included in boiler output but is shown on Form D to give the manufacturer an idea of the extent of this loss.

54. *Heat Loss from Uncovered Boiler* (ITEM *n*, Form A and B) is the heat lost from that part of the boiler surface not insulated during test which could be saved by the normal insulation of the boiler in practice and which would therefore appear as added output.

55. *Heat Loss from Uncovered Piping* (ITEM *o*, Form A and Form B) is the heat that is lost through uncovered piping (if condensation resulting drains back to the boiler and is re-evaporated) because the surfaces thereof are not normally insulated as specified in code. This correction is required only when piping cannot be insulated.

56. *Coefficient of Covered Boiler Surface* is the quantity of heat expressed in Btu transmitted from the steam and water in the boiler to the boiler room per sq ft of normal insulation per hour per degree difference in temperature between the steam and the boiler room.

57. *Actual Evaporation* is the actual weight of water heated from the average feed temperature to average steam temperature and evaporated into steam during a test.

58. *Factor of Evaporation* is the total number of Btu absorbed per pound of water evaporated divided by 971.7.

59. *Total Equivalent Evaporation* (ITEM *p*, Form A and Form B) is the quantity of water which would have been evaporated from a feed water temperature of 212 deg into steam at 212 deg by the corrected heat output delivered at the boiler nozzle.

60. *Boiler Output* is the quantity of heat available at the boiler nozzle with the boiler normally insulated.

61. *The Square Foot of Steam Radiation* is the heat output of 240 Btu per hour.

62. *A Damper* is a device used to control the flow of air or gases in a boiler.

63. *A Choke Damper* is a damper placed within the gas passage.

64. *A Check Damper* is a device for admitting air from the boiler room to the gas passage or smoke pipe.

Mr. Harding stated that the test codes were rather easy to handle, but the matter of the rating code was quite a different story and submitted a detailed report of the work of his Committee as follows:

Report of A.S.H.&V.E. Continuing Committee Codes for Testing and Rating Steam Heating Solid Fuel Boilers

THIS committee met in Buffalo, May 4, 1929, to tabulate and discuss the replies received to the questionnaire previously mailed to a list of boiler manufacturers and others interested in this matter. This questionnaire was printed in the April, 1929, issue of the JOURNAL of the A.S.H.&V.E. as follows:

CONTINUING COMMITTEE'S LETTER TO BOILER MANUFACTURERS

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

29 WEST THIRTY-NINTH STREET

NEW YORK

Feb. 11, 1929

As you are doubtless aware, the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at the Annual Meeting held in Chicago, January 28 to 31 of this year, adopted a code entitled, Code for the Rating of Heating Boilers Burning Solid Fuel.

The work of the original committee having been completed a continuing committee composed of the undersigned members was appointed by the President and charged with the duty of first determining the feasibility of applying this code in practice and secondly recommending to the Society such revisions as may develop after the criticism, opinion and suggestion of boiler manufacturers and other associations interested have been reviewed and given careful consideration.

We desire to call your attention to the fact that this code has been printed for discussion only and has not been released by the Society.

This letter is being mailed to the list of boiler manufacturers on file in the offices of the Secretaries of the A.S.H.&V.E., H.&P.C.N.A. and the N.B.&R.M.A.

The Committee is interested only in obtaining a uniform rating for boilers and will earnestly endeavor to make such recommendations to the Society as will be supported by the facts that may be developed.

We clearly recognize that the principle involved in any engineering code is primarily the protection of the public and incidentally the ultimate consumer who pays the bills. The material benefits of standardization in this case accruing to the manu-

facturer, engineer and contractor are self evident as amply proven by a multitude of existing examples.

We believe the industry involved earnestly desires a workable code which affords an equitable protection to the various interested parties concerned.

We desire to call your attention at this time to Section V of the A.S.H.&V.E. Code of Minimum Requirements for the Heating and Ventilating of Buildings. This code revised to date will be distributed by the Society at an early date.

There is no reference in this code to the subject of rating boilers. Section V as indicated by its heading, refers to a method to be employed in determining the load to be carried by the boiler for any steam or hot water installation. Any other method may of course be employed that will give equal or greater loads.

The following extract from the Minimum Requirements Code is given with the thought that it may have a bearing on the answers to some of the questions on which we desire information:

SECTION V

Minimum Capacity and Installation Requirements for Low Pressure Steam and Hot Water Heating Boilers

(1) *Estimated Boiler Load:* For the purpose of this Code the estimated connected load to the boiler or boilers employing solid fuels stated in Btu per hour shall be taken as the sum of the following items:

(*a*) The estimated heat emission in Btu per hour of the connected radiation, direct, indirect or both, to be installed as determined by computation from data given in Sections II and III for normal operation.

(*b*) The estimated maximum heat in Btu per hour required to supply water heaters or other apparatus to be connected to the boiler or boilers.

(*c*) The estimated heat loss in Btu per hour of the piping connecting radiation and other apparatus with the boiler or boilers (see Table I at the end of this section).

(*d*) The estimated increase in the normal load in Btu per hour due to starting with cold piping and radiation. This increase is to be based on the sum of items (*a*), (*b*) and (*c*) and shall be assumed not less than the following:

Sum of Items (*a*), (*b*) and (*c*) in Btu per hour			Percentage Increase to be added
Up to		100,000	65
100,000	to	200,000	60
200,000	to	600,000	55
600,000	to	1,200,000	50
1,200,000	to	1,800,000	45
Above		1,800,000	40

(2) *Boiler Capacity to be Installed:* The boiler or boilers to be installed shall be guaranteed by the manufacturer of the boiler to be capable of supplying, at the boiler outlets, the total Btu per hour as computed by the method outlined in the preceding paragraph and under the following conditions of operation each of which, is to be stated in the specifications covering the installation for which the boiler or boilers are intended.

The comments of the continuing committee on this portion of the Minimum Requirements Code follow:

It will be observed that the responsibility of correctly determining the maximum boiler output required as well as the output for the design load does not rest with the boiler manufacturer—unless, he also acts in the capacity of the engineer.

(1) It is evident that the boiler or boilers selected must have as a primary requisite sufficient maximum output to handle the starting up load (maximum demand) with the draft available from the chimney selected or recommended for the

proposed installation. The starting up load may occur frequently during the heating season but is naturally of comparatively short duration. The question of fuel economy (efficiency) is of relative unimportance for this load.

(2) The boiler load as determined from the heat emission of the radiation and its connecting piping (design load) in general exists for only comparatively short periods (few days) during the heating season.

The question of fuel economy (efficiency) is also in this case relatively unimportant.

(3) The average operating load on the boiler or boilers for the heating season will probably not exceed 35 per cent of the design load (2) in the average case.

Fuel economy (efficiency) for this average load is of considerable interest to the owner.

It will be observed that a small boiler, for example, must be capable of producing an output (with the chimney recommended) 65 per cent in excess of the design load output. The ratios of the three loads will be: 1.65 (max. load); 1.00 (design load); 0.35 (average seasonal load), the average load being approximately 21.2 percent of the maximum load.

In order that we may have your ideas and opinions on this subject, we have prepared the following short questionnaire:

1. Do you believe that a manufacturer should guarantee the listed ratings of the boilers he manufactures under conditions of operation he may state and dimensions of chimney he recommends?

2. Do you favor a "one number rating" for heating boilers? If so, what do you suggest as limiting factors to be employed by the manufacturer in determining the ratings listed?

Assuming the maximum load to which any boiler may be connected has been determined by the method as outlined by the A.S.H.&V.E. Code of Minimum Requirements previously given. What percentage of this load would you suggest the manufacturer employ in making up his published rating tables?

Do you recommend that the manufacturer state this percentage in his catalog?

Would you prefer in this case to recommend that the published rating be accompanied by a statement to the effect that the boiler (connected to the chimney listed) is capable of developing a certain percentage increase above the stated rating?

If, in your mind, the preceding suggestions do not appear practicable, how would you suggest this point be covered for the protection of the manufacturer as well as the engineer and contractor?

3. Do you favor a "multi number rating" for heating boilers? If so, what limiting factors do you suggest be employed by the manufacturer in determining the ratings listed?

Assuming that the maximum load to which any boiler may be connected has been determined by the method outlined by the A.S.H.&V.E. Code of Minimum Requirements previously given. What percentage of this load would you suggest the manufacturer employ in making up his published rating tables?

Do you recommend the manufacturer state this percentage with the rating tables? Would you prefer in this case to recommend that published ratings be accompanied by a statement to the effect that the boiler (connected to the chimney listed) is capable of developing a certain percentage increase above the stated ratings?

If, in your mind, the preceding suggestions do not appear practicable, how would you suggest this point be covered for the protection of the manufacturer as well as the engineer and contractor?

4. Do you prefer to see boiler output values stated in Btu per hour or in sq ft of "equivalent direct radiation" or both? If "equivalent direct radiation" is to be employed what Btu values would you assign to (*a*) steam radiation, (*b*) water radiation?

5. Please state all of the information that you believe should accompany the rating tables as published by the manufacturer such as:

Size, kind and Btu value of the fuel, grate area, dimensions of chimney, temperature of flue gas, rate of combustion, available fuel charge firing period, overall boiler and grate efficiency, etc.

6. Boiler manufacturers are requested to furnish this committee with output curves or performance tables for two boilers in each series they manufacture for use of solid fuels giving rate of combustion and smoke hood temperatures. What "one number rating" do you assign to these boilers?

If a 12 per cent CO_2 requirement was not met, please give the percentage you obtained in your output test.

Please acknowledge the receipt of this letter and advise if you are willing to co-operate by suggestions, etc., in the work of this committee in its attempt to coordinate the desires of the parties interested.

Please address your correspondence to the Society headquarters, 29 West Thirty-ninth St., New York, N. Y.

Yours very truly,

COMMITTEE ON INTERPRETATION OF CODE FOR RATING LOW-PRESSURE HEATING BOILERS
L. A. HARDING—*Chairman*
F. C. HOUGHTEN
R. V. FROST

Replies were received to the questionnaire from a large percentage of the membership of the *National Boiler and Radiator Manufacturers' Association*, some independent manufacturers of steel and cast iron heating boilers; also the following reply from the Boiler Output Committee of the *Heating and Piping Contractors National Association:*

Your letter of February 11, 1929, having reference to code adopted by the A.S.H.&V.E., at their annual meeting in January this year and having further reference to questionnaire prepared by a continuing committee charged with the duty of first determining the feasibility of applying this code in practice and looking for criticisms, opinions and suggestions from other interested associations:

Referring directly to the questionnaire prepared and replying to No. 1 will say—Yes, but the condition of operation should be standardized. In addition the manufacturer should publish the following—kind of fuel and Btu value, rate of combustion, firing period, attention period, flue gas temperature, stack draft, over-all efficiency, dimension of chimney and available fuel charge.

In answering the first paragraph of question No. 2, we are in favor of a No. 1 rating with a complete performance chart.

The second paragraph of question No. 2 has our same answer, prefer complete performance chart.

Third paragraph of question No. 2—our answer is "No."

Answering the 4th paragraph of question No. 2—published rating tables should have no bearing on determining loads so far as boiler manufacturers are concerned. Complete performance chart should be furnished.

Fifth paragraph of question No. 2—our answer is complete performance chart to be furnished.

Third question first paragraph—our answer is "No."

Second paragraph of question No. 3—our answer is, that manufacturer should not employ any percentage in his published rating tables.

This same applies to paragraphs No. 3 and No. 4 of question No. 3.

Answering question No. 4—we prefer to see Boiler Output values stated in direct radiation based on 225 Btu for steam and 150 Btu for water radiation.

In answering question No. 5 including first and second paragraphs, our answer is, that complete performance charts should be furnished, also actual grate dimension when measured by the *Heating and Piping Contractors National Association's* standard method, also outside dimensions of boiler, height of water line and depth of ashpit.

In answering question No. 6, our answer is, that all round boilers should be tested. Rectangular boilers, a sufficient number in each series, should be tested so that a reasonably accurate performance chart can be determined for the complete series of boilers.

We regret that the answers to your questions did not receive earlier attention but it was entirely overlooked until a wire was received from our main office in New York.

Yours very truly,
HEATING AND PIPING CONTRACTORS
NATIONAL ASSOCIATION
BOILER OUTPUT COMMITTEE
(Signed) GEORGE M. GETSCHOW, *Chairman*

The members of the heating division of the *American Boiler Manufacturers Association* did not, except in one instance, favor the committee with any replies to the questionnaire.

A summary of the replies received follows:

1. All are agreed that manufacturer should guarantee the listed ratings of the boilers manufactured under conditions of operation he may state and dimensions of chimneys given by him.

2. Several manufacturers apparently, favor a one number rating, one of which states that "it would be advisable to have a one number rating for price comparisons of boiler capacities, other capacities with performance characteristics are desirable."

Many of the answers given to this question are in effect, "we do not favor 'one number rating' with this exception, that, considering conditions in the industry, and this is entirely a commercial reason, the one number rating would go to some length to insure uniform price."

3. The great majority of the replies received indicate that a "multi-number rating" is favored.

The *Heating and Piping Contractors National Association* express a preference for complete performance charts.

4. Practically all prefer to state boiler outputs in both Btu per hour and sq ft of equivalent direct radiation.

5. The consensus of opinion in reference to the statement of conditions that should accompany rating tables are practically uniform on the following items:

(1) Combustion rate
(2) Firing period, attention or fuel available in hours
(3) Overall efficiency
(4) Draft tension
(5) Dimensions of chimney
(6) Fuel (kind not stated) 12,500 Btu per lb
(7) Grate area and chimney dimensions to be given
(8) The majority of the answers omit flue gas temperature as a condition. One suggests a limit of 800 F.

We believe that the thoughts of this committee and two associations previously mentioned, are apparently in harmony covering the principal features of a boiler rating code.

If a prospective purchaser is able to obtain from the literature of the manufacturer in advance of the actual purchase, guaranteed definite information of what he may

expect from the operation of the apparatus as to *output, economy* and *attention* under a specific set of conditions and over a reasonable range of operation, he will be provided with the necessary information required to select the size and apparatus to suit his particular condition and at the same time be able to predict the operating economy with reasonable accuracy which he may later expect to obtain.

This information can only be obtained through the medium of standardized boiler tests or trials.

It is obvious that no real progress could be expected in formulating a boiler rating code until a satisfactory boiler performance testing code be devised to give the information sought. Such a code is now in satisfactory operation by a number of the boiler manufacturers.

We believe the preceding paragraphs cover the principles involved on the subject of boiler rating.

The committee, after its meeting on May 4, formulated a tentative revision of the present A.S.H.&V.E. Code for Rating Low Pressure Heating Boilers, which was mailed to the same manufacturers and organizations that received the questionnaire. The proposed revision was accompanied by a copy of the committee's comments on the subject of boiler rating which follows:

GENERAL

A rating code for any type or kind of apparatus must of necessity include the following items:

1st—The output.

2nd—Specified conditions of operation for the output stated.

3rd—The limits placed on the specified conditions.

It is obviously necessary to standardize limits for some of the specified conditions, otherwise a rating code could not fulfill its function.

Some of the limits to conditions are naturally now set by custom and usage or by ordinances or laws, designed to protect the health or safety of the community.

For example: Custom and usage have decreed that the conditions relative to steam pressure for rating purposes shall be 2 lb gage at the boiler for heating boilers, whereas, overall efficiency, draft tension, temperature of flue gas and rate of combustion are conditions over which it is difficult if not altogether impractical to assign definite limits in the present state of this art with the possible exception of flue gas temperature. This should naturally be limited to a maximum value consistent with safety relative to fire hazard.

Placing a minimum value limit on boiler efficiency covering the average load period of the heating season, would appear to be in line with a program of fuel conservation and probably receive a welcome by prospective owners of heating boilers.

Not the least item, however, in boiler economy is the manner in which the boiler is actually operated by the owner and over which the manufacturer has no control.

The manufacturer can and does produce boilers which give relative high efficiency when properly operated but who can set a minimum efficiency limit that would produce the actual results that are apparently desirable.

There are a number of terms employed in this art, the meaning of which is either vague, not clear or by no means standardized. We give below the definition of various terms as we understand them.

Purchaser:

Construed to mean the person responsible for the selection of the boiler.

Equivalent Direct Radiation:

The heat emission of 240 Btu per hour per square foot of manufacturers' *rated surface* of direct steam radiation and 150 Btu per hour per square foot of manufacturers' *rated surface* of direct hot water radiation.

One Number Rating:

A single rating stated for each boiler listed in a manufacturer's catalogue.

Multi-Number Rating:

Two or more ratings stated for each boiler listed in a manufacturer's catalogue.

Heating Boiler Output:

As defined by the proposed A.S.H.&V.E. Performance Test Code for Steam Heating Solid Fuel Boilers.

Boiler Efficiency:

The overall efficiency of grate and boiler as defined by the proposed A.S.H.&V.E. Performance Test Code for Steam Heating Solid Fuel Boilers.

Estimated Maximum Load, Peak Load, Starting-Up Load:

These terms are considered synonymous and construed to mean the load, stated either in Btu per hour or equivalent direct radiation that has been determined by the purchaser to be the greatest estimated output that the boiler will be called upon to carry in operation. We recommend the use of the term estimated maximum load in this connection.

Net Load:

This term now has various meanings, all the way from the load of the radiation in normal operation to the maximum load. We recommend that this term be dropped from heating literature as serving no useful purpose.

Estimated Design Load:

The load, stated in Btu per hour or equivalent direct radiation, as estimated by the purchaser for the conditions of inside and outside temperature for which the amount of installed radiation was determined and is the sum of the heat emission of the radiation to be actually installed plus the allowance for the heat loss of the connecting piping plus the heat requirements for any apparatus connected with the system.

Estimated Average Load:

The estimated average load stated in Btu per hour or equivalent direct radiation for the heating season and based on the average outside temperature during the heating season for the locality in question.

Heating-Up Factor:

The percentage added to the estimated design load in determining the estimated maximum load.

Attention, Firing Period, Fuel Available in Hours:

The hours required to burn one available fuel charge. *Fuel available* is defined in the proposed performance boiler test code. We recommend the discontinuance of the use of the terms *attention* and *firing period.*

Load Rating Versus Output Rating:

These terms are not synonymous as now employed by various manufacturers. Some manufacturers prefer to include in their rating the allowance for the heat loss of connecting piping, or the heating-up factor or both, in either case the ratings stated are obviously not the boiler outputs required for either the estimated design load or the estimated maximum load, but are evidently some lower figure.

It does not appear to the committee that the boiler manufacturer should assume the responsibility of guaranteeing a rating based on the use of some heating-up factor of which the purchaser has no definite knowledge.

We believe the responsibility for the selection and application of a heating-up factor lies entirely with the purchaser just as much as it is his responsibility for determining the estimated design load. If he is capable of the latter he surely should be and is naturally capable of the former.

If a manufacturer includes in his literature the necessary heat transmission data for calculating the estimated design load he might very properly include a table of heating-up factors.

He does not guarantee the former so why should he guarantee the latter.

We do not see that the necessary process of selecting a boiler for a given load is

in any way simplified by including in the published rating designation allowances for various items of estimated value.

Boiler outputs are something quite tangible when determined by a uniform test code, whereas, boiler loads are only estimates.

If all ratings stated in catalogues were boiler outputs as determined by the provisions of the test code, the boilers of various manufacturers would evidently be more easily compared, which overcomes one of the greatest objections frequently referred to as the existing confusion in boiler ratings.

In the interest of uniformity in catalogue listing, this committee recommends that all ratings be stated in terms of boiler outputs. The terms, boiler load rating and boiler output rating, would be synonymous and we see no reason for employing the term boiler load rating.

Range and Number of Boiler Outputs Desirable per Boiler:

If the purchaser is to have any very definite idea of the economy of the boiler he proposes to install he should naturally be provided with the efficiency corresponding to the average output as determined by and equal to the estimated average load estimate. This is the lowest output that there would be any very good reason to list.

The estimated average load in localities where heating is important is frequently assumed to be approximately 35 per cent of the estimated design load.

If the heating-up factor is assumed as 65 per cent then the "estimated maximum load" becomes 165 per cent of the estimated design load and the estimated average load is 21.2 per cent of the "estimated maximum load."

It is recognized that these percentages are somewhat variable but it would appear reasonable to assume that the minimum output listed for a particular boiler should bear some relation to the maximum listed output and we suggest that approximately 20 to 25 per cent be employed in this connection. Thus, if 25 per cent was used the fuel available in hours for the minimum output would be 6.6 times that of the maximum output listed.

It is recommended that at least five outputs be given with the corresponding conditions for each boiler listed in order that those who are interested in the use of boiler performance charts may have the desired information to plot such charts. This would require listing three outputs fairly equally spaced between the minimum and maximum.

"Fuel Available in Hours" for Maximum Output:

If 8 hours be considered a desirable time as set by custom and usage for the smaller size boilers to employ for the design load, then on the basis of a 65 per cent heating-up factor the fuel available in hours for maximum output becomes 4.84 hours. If 6 hours is satisfactory for the larger boilers to employ for the design load the fuel available in hours for maximum output becomes 3.7 hours.

In the interest of uniformity in rating it would be well to place some limits on the fuel available in hours for the maximum outputs listed, although we have no recommendations to make at this time.

Flue Gas Temperature:

It is coming to be more generally recognized that flue gas temperature *is not* a reliable indicator of efficiency and in many cases only remotely connected with efficiency.

The information in reference to both the temperatures of flue gas and draft tension is, we believe necessary, particularly for boilers of the larger size.

The purchaser must have this information if he is to be able to perform the necessary calculations in determining satisfactory chimney or stack dimensions.

This is more particularly true for boiler installations serving high buildings.

If only draft tension is stated the purchaser is left to make his own estimate as to stack temperature. The manufacturer should be as much concerned in the satisfactory operation of the installation as the purchaser and this is hardly possible unless a proper size chimney or stack is provided by the purchaser.

Boiler Test Codes:

It is proposed to recommend perhaps several minor changes in several items of the A.S.H.&V.E. Code for Testing Low Pressure Steam Heating Solid Fuel Boilers.

We also propose to recommend to the Society the adoption of the test code now employed by the *National Boiler and Radiator Manufacturers Association* and to be designated as the A.S.H.&V.E. Performance Test Code for Steam Heating Solid Fuel Boilers.

This Committee met with representatives from the *American Boiler Manufacturers Association* and the *National Boiler and Radiator Manufacturers Association* in Buffalo, June 20, 1929, for the purpose of discussing the proposed revision of the present A.S.H.&V.E. Code for Rating Low-Pressure Heating Boilers.

As a result of this conference your Committee and the representatives from the *National Boiler and Radiator Manufacturers Association* agreed upon the following draft of a rating code. The representatives present at this meeting from the *American Boiler Manufacturers Association* did not agree with this code. Their views on the question of rating heating boilers are expressed by the letter which follows:

We, the undersigned manufacturers of low pressure steel heating boilers, representing approximately 95 per cent of the total steel heating boiler industry, object to the Code for rating low pressure heating boilers as proposed by your Committee, dated May 17, 1929. The objections are based on the following reasons:

1. The selection of boilers as based on the maximum capacity, would result in smaller steel boilers being used on a specific job as compared to present practice, with a consequent lowering of operating efficiency.

This Code specifies minimum requirements, and it has been experienced that in similar Codes, where minimum requirements are given, minimum requirements are used.

The present practice in selecting steel boilers gives a uniformly higher reserve capacity than as proposed by the Code.

2. The proposed Code does not consider heating surface, which is a prime essential in determining boiler capacity or ratings, and on which basis steel heating boilers have been rated and satisfactorily selected for a great number of years.

3. We object to the use of a multi-rating, as it would tend to cause confusion in an industry that has used but a single rating satisfactorily for a number of years.

FOR TESTING AND RATING STEAM HEATING SOLID FUEL BOILER

4. We object to the use of anthracite fuel only as a basis for determining rating.

Our recommendations regarding the selection and rating of heating boilers are based on a practice which has proven to be satisfactory to both Manufacturer and User throughout the entire life of the steel heating boiler industry, and are as follows:

(1) The published boiler rating shall correspond to the "estimated design load" as defined in *Section* 5 of the CODE OF MINIMUM REQUIREMENTS, being the sum of Items A, B and C in Paragraph 1.

(2) The published boiler rating shall be determined by the amount of heating surface and grate area.

We are prepared to furnish your Committee with exact information as to minimum ratios of rating to heating surface and heating surface to grate area to be employed in determining the rating for heating boilers.

Yours truly,

KEWANEE BOILER CORP.
PACIFIC STEEL BOILER CORP.
HEGGIE-SIMPLEX BOILER CO.
OIL CITY BOILER WORKS
INTERNATIONAL BOILER WORKS CO.
TITUSVILLE IRON WORKS CO.
FITZGIBBONS BOILER CO., INC.
AMES IRON WORKS
THE BROWNELL CO.
ERIE CITY IRON WORKS

Your Committee feels that they have secured the support of a large percentage of the manufacturers of heating boilers and submits the following revision of the present boiler rating code for your consideration.

Proposed Revision of January, 1929 A.S.H.&V.E. Code for Rating Steam Heating Solid Fuel Boilers

(1) PURPOSE

The purpose of this Code is to standardize the method to be employed and followed by any person, partnership, firm, corporation or association, who may desire to make use of, or employ for any purpose whatsoever the statement "The rating of the boilers herein listed are in accordance with the provisions of the A.S.H.&V.E. Code (year) for Rating Steam Heating Boilers Burning Solid Fuel."

(2) RATING DESIGNATION

It is understood that all ratings stated are boiler outputs for the corresponding boiler designation as were determined and defined by the provision of the "A.S.H.&V.E. Performance Test Code for Steam Heating Solid Fuel Boilers (year)" and as governed by the conditions as set forth under paragraph 4 and accompanying the ratings.

The output for each boiler shall be stated in thousands of Btu per hour and also in square feet of equivalent direct radiation. It shall be optional to state, in addition to the two methods indicated, the output in pounds of steam per hour.

(3) RANGE OF OUTPUTS FOR EACH BOILER LISTED

There shall be stated a minimum number of five boiler outputs for each boiler listed. The outputs shall have a range from maximum output to approximately 35 per cent of the maximum output and the intermediate outputs given are to be approximately equally spaced between the minimum and maximum outputs.

(4) LIST OF CONDITIONS, STATEMENTS OF LIMITING CONDITIONS AND MANUFACTURER'S GUARANTEE

There shall be stated under each output listed the numerical values for each of the following five items:

1. Fuel available in hours.
2. Combustion rate pounds per hour per square foot of grate surface.
3. Overall efficiency per cent.
4. Average draft tension, inches water.
5. Interior dimensions of chimney and height.

The following statements shall be included under each table of rating.

"The *priming* for any output listed above does not exceed two (2) per cent."

For Anthracite Fuel

"The ratings are based on a steam pressure of 2 lb gage at the boiler and anthracite coal stove size, having a calorific value of 12,500 Btu per lb on a moisture free basis."

For Bituminous Fuel

"The ratings listed are based on a steam pressure of 2 lb gage at the boiler and Bituminous Coal 3 in. by 2 in. size, having a calorific value of 13,000 Btu per lb sulphur content not exceeding 2 per cent and volatile content of not less than 30 per cent on a moisture free basis."

For Coke Fuel

"The ratings listed are based on a steam pressure of 2 lb gage at the boiler and by-product or gas coke of commercial size best suited to the boiler."

"The inside dimensions and height of chimneys listed should be satisfactory when properly constructed and having no other opening except for the purpose of serving the boiler and when free from the effect of adverse air currents. Allowance should be made for any other chimney openings, elbows in the smoke flue or breeching and for extra long smoke flue or breeching."

(5) TABLE OF DIMENSIONS

A comprehensive table of dimensions of the boilers listed shall be included in the same bulletin or catalogue with the ratings. This table shall include the number and pipe size of steam and return connections and location, smoke flue dimensions and height above floor line, grate area and height of boiler water line and such other dimensions as may be required for properly indicating the boiler to scale on a set of complete heating installation plans.

(6) DEFINITIONS

Purchaser:

Construed to mean the person responsible for the selection of the boiler.

Manufacturer:

The individual, firm or corporation who manufactures the boilers for which corresponding ratings are listed.

Boiler Output:

As defined by the proposed A.S.H.&V.E. Performance Test Code for Steam Heating Solid Fuel Boilers.

Estimated Maximum Load:

Construed to mean the load stated in Btu per hour or equivalent direct radiation that has been estimated by the purchaser to be the greatest or maximum load that the boiler will be called upon to carry.

Equivalent Direct Radiation:

Construed to mean the heat emission of 240 Btu per sq ft of manufacturer's rated surface of direct steam radiation and 150 Btu per hour per sq ft of manufacturer's rated surface of direct hot water radiation.

Grate Area:

As defined by the A.S.H.&V.E. Code for Testing Low Pressure Steam Heating Solid Fuel Boilers.

Fuel Available in Hours:

Construed to mean the hours required to burn one available fuel charge. The available fuel is defined by the proposed A.S.H.&V.E. Performance Test Code for Steam Heating Solid Fuel Boilers.

Overall Efficiency:

As defined by the proposed A.S.H.&V.E. Performance Test Code for Steam Heating Solid Fuel Boilers.

Priming:

The amount of free moisture carried by the dry saturated steam vapor delivered by the boiler stated as a percentage of the total weight of the sum of the dry saturated steam plus the free moisture delivered.

Signed:

L. A. Harding, *Chairman*

R. V. Frost F. C. Houghten

A.S.H.&V.E. Continuing Committee Codes for Testing and Rating Steam Heating Solid Fuel Boilers

In summarizing the results of the Committee's work, Mr. Harding stated that the proposed revision of the January, 1929, A.S.H.&V.E. Code for Rating Steam Heating Solid Fuel Boilers simply requires a complete picture of boiler performance on the basis of output rating and from the data derived, the purchaser can draw his own performance curves, calculate his loss from the struc-

ture and apply his own heating-up factor. The Committee does not bring this report for approval or adoption but merely for discussion, he said.

The report of the Committee on the Code for Heating and Ventilating Garages was submitted by W. H. Carrier, a member of the Committee, in the absence of E. K. Campbell, Kansas City, chairman, who was injured in an automobile accident enroute to the convention. Upon motion by W. T. Jones, the report was unanimously accepted and adopted as a code of the Society.

Report of Committee on Heating and Ventilating Garages

In May, 1928, the *National Fire Protection Association* adopted tentative regulations governing the construction and operation of bus garages prepared by its Committee on Garages, which had the following personnel: H. E. Newell, Chairman; K. W. Adkins, F. H. Alcott, E. P. Boone, K. B. Brier, E. K. Campbell, A. M. Daniels, W. K. Estep, R. H. Goodwin, Louis Harding, G. C. Hecker, W. F. Hickey, A. D. Knox, R. C. Loughead, Ray Nelson, I. Osgood, R. E. Plimpton, A. M. Schoen, H. S. Smith, John Stilwell, J. F. Templin, J. S. Trump, W. B. White and Wm. P. Yant.

An abstract from the code consisting of Sections 15 to 19, prepared jointly by a committee of the Society consisting of E. K. Campbell, chairman; W. H. Carrier, E. B. Langenberg and Thornton Lewis, appears in the following paragraphs:

Section 15.

(a) Heat generating plants should preferably be located in a detached building. If within the garage, the heat generating plant shall be placed in a separate room used for no other purpose and cut off horizontally and vertically from all other parts of the building by reinforced concrete walls not less than 6 in. thick, or masonry walls not less than 8 in. thick.

Openings in such walls shall be restricted to those necessary for heating pipes and ducts, which openings shall be made tight in a manner to provide for expansion and at the same time prevent air passing through the walls. Entrance to room containing the firing space shall be from the outside only.

All air entering the heat generating plant for combustion purposes shall be drawn from outside the building.

(b) Sufficient heating capacity shall be provided to maintain an inside temperature of not less than 40 F in the coldest weather and to maintain a temperature of not less than 35 deg at the warm air inlet when the ventilating system is in full operation for flushing purposes.

(c) No method of heating shall be used which permits fire in the garage or in any communicating room.

(d) Motors used in connection with heating system shall be of the constant speed type. All switches and motors shall be of approved design and installed in compliance with the National Electrical Code. Polyphase motors shall be protected against single phase operation.

(e) The use of steam or hot water heating systems by either direct or indirect radiation is permitted, provided the requirements of the ventilating section of these regulations are complied with. Inside air inlets for indirect radiation shall not be higher than 2 ft above the floor.

(f) Unit heaters employing steam or hot water are permitted provided the requirements of the ventilating section of these regulations are complied with.

(g) Steam blast systems with central fan and coils together with ducts are permitted provided the requirements of the ventilating section of these regulations are complied with. The heating coils of such systems shall be separated from the firing space by masonry walls at least 8 in. thick.

(h) Warm air furnace blast systems of heating are permitted provided the requirements of the ventilating section of these regulations are complied with. The air space surrounding the furnace within the heating chamber shall be separated from the

firing space by a masonry wall at least 8 in. thick or other wall of material and thickness which may be approved by the Underwriters. This wall may be formed in part by the furnace front which must be not less than ¼ in. thick, of steel or cast iron and this furnace front may be pierced only by the feed and ash pouches of the furnace, and by clean-out doors opening into the combustion space or smoke passages. Access doors through either brick wall or furnace front shall not be permitted. Warm air furnaces other than blast systems shall not be permitted.

(i) Positive recirculation of 1 cu ft of air per sq ft of floor area within the garage shall be provided in all garages having an average ceiling height of not more than 15 ft and in all garages having greater ceiling height than 15 ft the volume of air recirculated shall be increased in proportion to the increase in height. The return air openings in such recirculating systems shall be not more than 2 ft above the floor.

(j) In central furnace fan plants, not less than 5 per cent of the air moved by the fan shall be taken direct from outside of the building through a duct which shall deliver the outside air to a point near the floor on which the fan rests; the duct shall be open at all times and the air supply which is provided shall be without control.

(k) All fans used for recirculating air within the garage or exhausting air from the garage shall be of non-sparking type.

Section 16. Ventilation of Storage Sections.

(a) These regulations shall apply to the following garages:

1. Garages housing 35 or more motor vehicles with 3 or more walls pierced with openings.

2. Garages housing 25 or more motor vehicles with 2 walls pierced with openings.

3. Garages housing 4 or more motor vehicles and located above ground, but having less than 2 walls pierced by openings and exposed to the outside.

4. Garages housing 4 or more motor vehicles and located below the level of the ground.

(b) Natural ventilating may be employed where it is practicable to maintain open windows or other openings at all times. Such openings shall be distributed as uniformly as possible in at least two outside walls. The total area of such openings shall be equivalent to at least 5 per cent of the floor area.

(c) Where it is impracticable to operate such a system of natural ventilation, a mechanical system of ventilation shall be provided. This system may be combined with the heating system or may be an entirely separate installation.

(d) Positive provision shall be made for either the inlet of 1 cu ft of air per minute from out-of-doors for each square foot of floor area, or for removing the same amount and discharging it to the outside as a means of flushing the garage.

(e) For the purpose of these regulations, positive means of handling air shall be understood to mean a power-driven fan or fans of sufficient capacity to move the required volume of air.

(f) Where positive systems of exhausting air are used, the exhaust openings shall be not more than 24 in. above the floor and shall be not more than 50 ft apart.

(g) Garages having a capacity of not less than 4 or not more than 35 cars within the scope of these regulations may consider air exhaust stacks as positive, provided they have not less than 15 sq ft of steam heating surface for each square foot of duct area, and not less than 1 sq ft of free area through both heating coil and duct for each 350 sq ft of floor area. Such an exhaust duct shall discharge above the roof and extend in any case to a height of not less than 15 ft above the heating coils.

(h) Where mechanical systems of introducing outside air are used, and where air is recirculated the air shall be delivered horizontally and in sufficient volume and with sufficient velocity to secure distribution to all parts of the building. The height of the air inlet opening shall be such that the air will be discharged above the top of the vehicles.

(i) All duct openings, either supply or exhaust, shall be covered with ¼-in. mesh screen.

(j) The passing of air ducts through fire walls shall be avoided wherever possible. Ducts shall be installed in accordance with the regulations for the installation of blower and exhaust systems.

Section 17. Repair Shops.

(a) Repair shops shall be ventilated as required for garage storage sections, except that mechanical means shall be provided for both the inlet and exhaust of 1 cu ft of air per minute per square foot of floor area.

(b) In connection with engine testing it is recommended that the engine discharge direct to outdoors through a straight duct or pipe of incombustible material, and of suitable size, installed as an extension of the exhaust pipe or muffler, in which case the mechanical system for inlet or mechanical system for exhaust may be omitted.

Section 18. Fuel Burning Appliances.

(a) Steam generators for tire vulcanizing, for oil and grease removal and for purposes other than space heating water heaters, and other fuel burning appliances such as forges shall not be installed within bus operating section or within carpenter or paint shop.

Section 19. Inspection and Repair Pits and Trestles.

(a) Elevated trestles or hoists are preferable for this service. If pits are used, they shall be continuously ventilated by a system independent of the main garage ventilating system. Such pits shall be cleaned at least daily and no accumulation of oil and grease permitted. Permanent illumination shall be provided.

J. R. McColl, Detroit, stated that the news of the recent death of E. S. Hallett, St. Louis, came as a distinct shock to everyone who knew him and that resolutions had been prepared and adopted by the Council which he thought should be presented to the meeting. The following resolutions were unanimously passed and an engrossed copy was to be sent to Mr. Hallett's family.

Whereas, Edwin S. Hallett, an Active Member of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, passed out of this life on March 29, 1929 and

Whereas, he was one of the Society's most loyal members, constant in attendance, contributing to the interest and value of the meetings by presenting his original ideas, and

Whereas, being widely known as an outstanding engineer in school house heating and ventilation practice, and

Whereas, the Society will miss his genial personality, his words of cheer and encouragement, his personal interest in each member and his kindly expressions of friendship, therefore, be it

Resolved, that we, the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, deeply regret the passing of our fellow member, Edwin S. Hallett, that we cherish the memory of his frank, upright character and that we hold in loving admiration his qualities as an independent and courageous thinker, and also, be it

Resolved, that a copy of these resolutions be spread upon the records of the Society and that a copy be sent to his family.

President Lewis announced the appointment of a Committee on Resolutions with the following personnel: W. H. Driscoll, Chairman, H. P. Gant and C. L. Riley, who submitted the following:

Resolved, that the congratulations of this Society be extended to the members of the Ontario Chapter, for the thoroughness with which they have attended to the many details necessary to the carrying out of a meeting with such a high measure of success as has attended their efforts here; and be it further

Resolved, that the thanks of the Society be conveyed to the Ontario members for providing a meeting place so unique and so delightful, and for a program of entertainment and sports that has so happily provided for the pleasure of every visiting member and guest.

Resolved, that this Society express its appreciation to the charming women of Ontario who served so splendidly as members of the Ladies' Committee, and whose presence helped make this meeting the great success that it is; and that the Society express its regrets that Mrs. Arthur S. Leitch, the Convener of the Ladies' Committee should have been unable to be with us because of the sudden illness of her daughter, and express its hope for the speedy recovery of the young lady.

Resolved, that it is the sense of this meeting that the Ontario Chapter could have made no better selection for the place of this meeting than Bigwin Inn; and that we express our appreciation to the management and staff of the Inn, who, confronted with the unusual situation of having to open up in advance of the regular season, for our accommodation, have met our needs so skillfully and satisfactorily.

Resolved, that by popular acclamation this gathering place itself on record expressing its admiration for the amazing skill exhibited by the Skipper of the Toonerville Trolley in conveying such a large number of our members and guests in perfect safety over the hazardous and tortuous passage of the Portage.

W. H. Driscoll presented the following resolution which was unanimously adopted:

Whereas, THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS is assembled in its 35th Semi-Annual Meeting, and

Whereas, for the first time in its history, the Society is meeting outside the limits of the United States, and

Whereas, this most successful and delightful event is being held under the auspices of the Ontario Chapter and within the confines of the British Empire; therefore,

Be It Resolved, that the greetings of this Society be extended to the members of the *Institution of Heating and Ventilating Engineers of Great Britain;* and

Be It Further Resolved, that as an additional mark of our esteem these Greetings be personally conveyed and delivered to the President of the *British Institution of Heating and Ventilating Engineers* by the President of this Society, Mr. Thornton Lewis.

Report of Exposition Committee

TO THE COUNCIL,
AMERICAN SOCIETY OF HEATING & VENTILATING ENGINEERS,

Your Committee on the International Heating and Ventilating Exposition wishes to report that under the able management of the International Exposition Co., plans for the International Heating and Ventilating Exposition are progressing rapidly, and this exposition should prove highly successful from every standpoint.

Definite arrangements have been made to hold the exposition in the Commercial Museum, in Philadelphia, from January 27 to 31, 1930, inclusive.

The Commercial Museum is admirably fitted for the holding of such an exposition, as all exhibits will be on one floor, which contains approximately 100,000 sq ft of space.

To date, definite contracts have been signed with 120 exhibitors and the space so taken represents approximately 40 per cent of the total space available.

With the exposition still six months in the future, it is believed that no difficulty will be experienced in contracting for space, and it is further believed that every phase of heating and ventilating industry will be represented at this exposition.

Great care has been exercised by your Committee and the International Exposition Co., to eliminate all companies whose products do not directly bear on heating and ventilating work.

The members of the Society can do much toward making this exposition a definite success, by urging heating and ventilating manufacturers to use space and exhibit their products.

Respectfully submitted by ADVISORY COMMITTEE OF INTERNATIONAL HEATING & VENTILATING EXPOSITION.

June 28, 1929: H. P. GANT, *Chairman*

PROGRAM

Wednesday, June 26

8:30 a.m.—Registration.

9:30 a.m.—Greeting by Ontario Chapter President.
Response by Pres. Thornton Lewis.
Instruments for the Measurement of Air Velocity, by Prof. J. H. Parkin.
Analysis of the Over-all Efficiency of a Residence Heated by Warm Air, by Prof. A. P. Kratz and J. F. Quereau.
Air Conditioning System of a Detroit Office Building, by H. L. Walton and L. L. Smith.
Heat and Air Volume Output of Unit Heaters, by Prof. L. S. O'Bannon.
Report of Committee on Code for Testing and Rating Unit Heaters, by D. E. French, Chairman.

Thursday, June 27

9:30 a.m.—Errors in the Measurement of the Temperature of Flue Gases, by Percy Nicholls and W. E. Rice.
Pipe Sizes for Hot Water Heating Systems, by Prof. F. E. Giesecke and E. G. Smith.
Report of Guide Publication Committee, by S. R. Lewis, Chairman.
Capacity of Radiator Supply Branches for One and Two-Pipe Systems, by F. C. Houghten, M. E. O'Connell and Carl Gutberlet.
Report of Committees on Interpretation of Code for Rating Low Pressure Heating Boilers, by L. A. Harding, Chairman.

Friday, June 28

9:30 a.m.—Determining the Quantity of Dust in Air by Impingement, by Prof. F. B. Rowley and John Beal.
Five Suggested Methods of Appraising Insulations, by Paul D. Close.
Time Lag as a Factor in Heating Engineering Practice, by James Govan.
Over-all Heat Transmission Coefficients Obtained by Tests and by Calculation, by F. B. Rowley, A. B. Algren and J. L. Blackshaw.
Report of Committee on Code for Heating and Ventilating Garages, by E. K. Campbell, Chairman.
Report of Advisory Committee on Heating and Ventilating Exposition, by H. P. Gant, Chairman.

ENTERTAINMENT EVENTS

Tuesday, June 25

12:00 a.m.—Special train leaves Toronto for Bigwin Inn via Huntsville.
6:00 p.m.—Arrive Bigwin Inn—Dinner from 6:30 to 8:00 p.m.
8:15 p.m.—Special program for Ladies—Introduction of Chapter Officers in the Rotunda.
9:00 p.m.—Informal reception and dancing in the pavilion. Motion pictures in the Rotunda.

Wednesday, June 26

8:30 a.m.—Registration in the pavilion.
10:30 a.m.—Special hiking party for ladies to explore Bigwin Island.
1:30 p.m.—Special Boat trip around Bigwin Island and Lake-of-Bays for ladies.
2:30 p.m.—Qualifying Golf Match.
7:00 p.m.—Semi-Annual Banquet in main dining room followed by dancing.

Thursday, June 27

9:30 a.m.—Golf tournament for ladies.

2:30 p.m.—Bridge party and tea for ladies.

2:30 p.m.—Golf tournament for Research Cup.

8:30 p.m.—Amateur Theatricals by cast of Ontario Chapter's members followed by Masquerade Ball in the Ballroom.

Friday, June 28

2:00 p.m.—Golf Match.

Tennis Tournament.

Lawn Bowling.

International Bang-and-Go Back Motor Boat Racing Contest (each Chapter will have a boat).

Log Rolling Contest by local talent.

COMMITTEE ON ARRANGEMENTS

M. BARRY WATSON, *General Chairman*
ARTHUR S. LEITCH, *Entertainment*
M. F. THOMAS, *Reception*
A. J. DICKEY, *Finance*
E. B. SHEFFIELD, *Transportation*
E. M. DOLAN, *Publicity*

LADIES COMMITTEE ON ARRANGEMENTS

MRS. A. S. LEITCH, *Convener*
MRS. M. F. THOMAS
MRS. E. B. SHEFFIELD
MRS. H. R. FLETT
MRS. M. B. WATSON
MRS. A. J. DICKEY
MRS. R. W. M. MCHENRY
MRS. H. H. ANGUS

No. 836

ANALYSIS OF THE OVER-ALL EFFICIENCY OF A RESIDENCE HEATED BY WARM AIR

By A. P. Kratz[1] (*Member*) and J. F. Quereau[2] (*Non-Member*), Urbana, Ill.

Acknowledgments

THE data presented in this paper were obtained in connection with an investigation which is being conducted by the Engineering Experiment Station of the University of Illinois, of which M. S. Ketchum, Dean of the College of Engineering, is Director, in co-operation with the *National Warm Air Heating Association,* under the supervision of A. C. Willard, Professor of Heating and Ventilation and Head of the Department of Mechanical Engineering. The basic data from which this paper is prepared are given in Engineering Experiment Station Bulletin No. 189.[3]

Under the terms of a co-operative agreement between the *National Warm Air Heating Association* and the University of Illinois, a very extensive study of furnace heating problems has been made, using first an experimental plant with auxiliary equipment in the laboratory, and later a typical modern residence erected by the association for the express purpose of correlating and extending the work in the laboratory to the conditions of actual installation. It was in this residence that the data herein presented were obtained.

Introduction

The design of house heating plants has usually been based on the assumption that the only heat available for compensating for the heat loss from the house would be that actually delivered to the rooms by the heating plant. Thus a boiler or furnace operating at 60 per cent efficiency would be expected to utilize 60 per cent of the heat of the fuel burned and to deliver this heat at the boiler nozzle or furnace bonnet. The loss between the heating unit and the rooms would then be deducted and only the remainder, usually less than 50 per cent of the heat of the fuel, would be regarded as available for actually heating the rooms.

Data obtained in the warm air heating research residence have indicated that in a self contained heating system much of the assumed waste heat is available for heating the house and is utilized quite efficiently. With efficient combustion,

[1] Research Professor, Engineering Experiment Station, University of Illinois.
[2] Special Research Assistant, Engineering Experiment Station, University of Illinois.
[3] "Investigation of Warm-Air Furnaces and Heating Systems, Part IV, Research Residence," by A. C. Willard, A. P. Kratz and V. S. Day, Engineering Experiment Station Bulletin No. 189.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Bigwin Inn, Lake-of-Bays, Ontario, Canada, June, 1929.

most of the 50 to 60 per cent usually assumed as loss, is used for heating the house and results in a very high over-all house efficiency. It is with this over-all house efficiency that this paper is concerned.

DESCRIPTION OF RESIDENCE

In all respects, the research residence, Fig. 1, is of standard frame dwelling construction, with the single exception of the studding, which is 2 in. × 6 in. instead of the usual 2 in. × 4 in. This permits the use of larger wall stacks, or vertical heat pipes, than could be used in 2 in. × 4 in. construction. The wall section is as follows: weather boarding, building paper, ship-lap siding on 2 in. × 6 in. studding, lath, and plaster with rough sand finish. The coefficient of heat transmission for this wall section is 0.20 Btu per square foot per hour per degree Fahrenheit, at a wind velocity of 15 mph. The walls are not

FIG. 1. WARM-AIR HEATING RESEARCH RESIDENCE, AT URBANA, ILL.

insulated, and no weatherstripping is used at the windows and doors. Interlocking copper shingles are used on the roof. The research residence has not been occupied by a family. A caretaker has lived in the residence, and the daily occupants have been the members of the research staff. Furniture, rugs, and window shades and curtains were provided. No cooking or other domestic activities requiring the application of heat were carried on. Thus the kitchen, as well as all other rooms, received heat solely from the heating system.

The room arrangement and exposures are shown in Figs. 2, 3, and 4. It should be noted that only one room, the bath-room, has a single exposure to the weather, and that throughout the residence the proportion of glass area is high. Hence the heating problem was typical of residence heating. The heat losses calculated by the standard methods are approximately 119,000 Btu per hour at 0 F, and 15 mph wind velocity.

HEATING PLANT

A gravity circulating warm-air heating plant was installed, and has been in operation in the residence continuously during each heating season since December, 1924. The heater was of a common cast-iron type, and had a grate

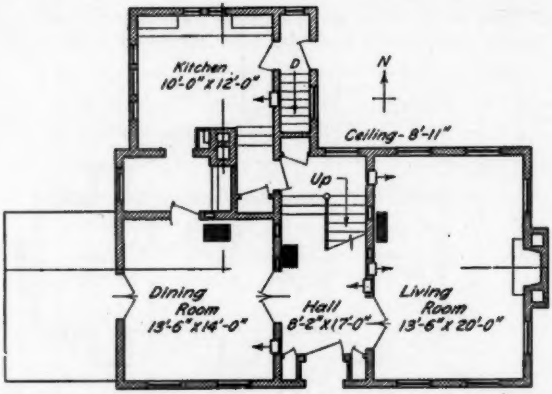


FIG. 2. FIRST FLOOR PLAN WITH LOCATION OF WARM-AIR REGISTERS AND COLD-AIR RETURN GRILLES (FOR THIRD INSTALLATION)

area of 2.88 sq ft. The smoke pipe, 10 in. in diameter and 10 ft in length, was connected to a 12-in. × 12-in. fireclay-lined flue, which was 35 ft high. This chimney had 8-in. brick walls, and passed up through the house. A cross damper in the smoke pipe, 3 ft from the furnace, was used to restrict the draft. The check draft was sealed.

The location of the warm-air registers is shown in Figs. 2, 3 and 4, and details of the piping and registers are given in Table 1. A single cold air return was in use when the data herein presented were obtained. The cold air was returned through a 36-in. × 36-in. square wood grille and 33-in. round duct, from a position in the hall near the main entrance of the residence. The free area of the grille was 800 sq in. and the duct area 854 sq in.

The heat pipes and fittings, eleven in number, were of standard commercial

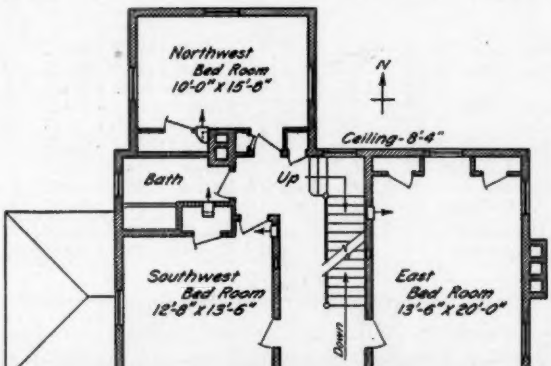


FIG. 3. SECOND FLOOR PLAN WITH LOCATION OF WARM-AIR REGISTERS

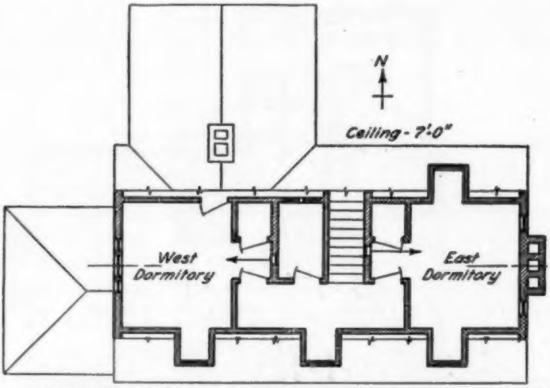


FIG. 4. THIRD FLOOR PLAN WITH LOCATION OF WARM-AIR REGISTERS

sizes and types, no effort being made to obtain streamline flow by the use of special fittings. All pipes were of bare bright tin, except for narrow sealing strips of asbestos paper at the joints, previous tests having demonstrated that asbestos paper covering on bright tin pipes is wasteful of heat.[4] The wall stacks were of mixed construction, some being of double tin with intervening air space, and others of single tin construction. The cross-sectional area of the stacks averaged 70 per cent of the area of the basement heat pipes to which they were connected. Registers were of commercial types and sizes and, in this particular installation, were all of the wall types.

TABLE 1. LEADER PIPE, STACK, AND REGISTER SIZES USED IN WARM AIR RESEARCH RESIDENCE

Story	Room	Leader		Stacks (or Throats)			Registers		
		Dia. In.	Area Sq In.	Size In.	Type	Area Sq In.	Size In.	Free Area Sq In.	Free Area Per Cent of Total
First	Living Room, N.	10	78.5	5½x13	Double	71.5	10x12	83.5	70
	Living Room, S.	10	78.5	5½x13	Double	71.5	10x12	83.5	70
	Hall	12	113.0	7 9/16x14	Single	106.0	12x14	120.5	72
	Dining Room	10	78.5	5½x13	Double	71.5	10x12	83.5	70
	Kitchen	12	113.0	7 x14	Single	98.0	12x14	120.5	72
Second	E. Bedroom	10	78.5	5 x12	Single	60.0	10x12	83.5	70
	S. W. Bedroom	9	64.0	3½x12	Single	42.0	9x12	74.0	69
	Bathroom	8	50.0	3 x10	Double	30.0	8x10	53.0	67
	N. W. Bedroom	10	78.5	5½x13	Double	71.5	10x12	83.5	70
Third	E. Dormitory	8	50.0	3 x10	Single	30.0	8x10	53.0	67
	W. Dormitory	8	50.0	3 x10	Double	30.0	8x10	53.0	67
TOTALS									
First			461.5			417.5			
Second			271.0			203.5			
Third			100.0			60.0			
Total			832.5			681.0			

[4] Emissivity of Heat from Various Surfaces, by V. S. Day, University of Illinois Experiment Station Bulletin No. 117.

It has been the object in this description of the research residence and the heating plant to show that both were standard rather than special, and that,

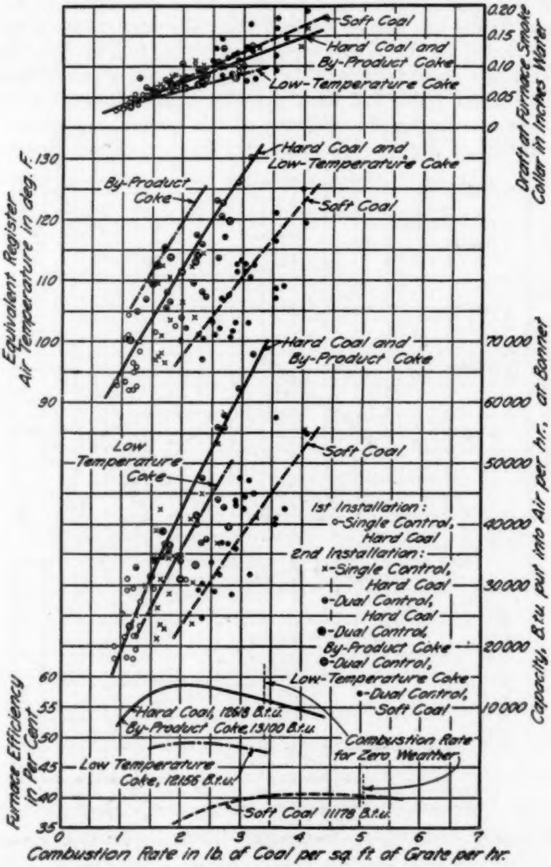


FIG. 5. PERFORMANCE CURVES FOR HARD AND SOFT COAL AND COKE

therefore, the data which follow are such as might be obtained in any well designed warm-air heating plant.

FURNACE PERFORMANCE WITH HARD AND SOFT COAL

The performance of the plant with anthracite coal, low-temperature and by-product coke, and bituminous coal is shown by the curves in Fig. 5.

With anthracite coal, the maximum efficiency of the furnace proper was 58.5 per cent, and the corresponding combustion rate was approximately 2.5 lb coal

burned per square foot of grate per hour. With soft coal, the maximum efficiency was only 41 per cent and occurred at a combustion rate of over 4 lb.

Of the heat in the air at the furnace bonnet (which at a maximum was 58.5 per cent of the heat of the fuel burned), only 75 per cent was delivered at the registers. This is shown by the curves of Fig. 6. This loss of heat between the bonnet and registers consists of heat loss from the leaders in the basement and from the stacks in the walls, and takes place by both radiation and convection. It has the effect of reducing the efficiency of the system as a whole, as shown in the lower curves of Fig. 6. Here the efficiencies as of the bonnet and as of

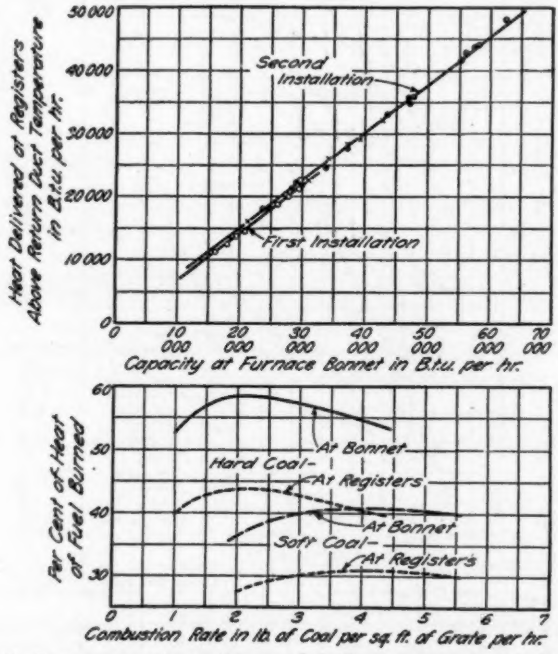


FIG. 6. LEADER AND STACK LOSSES, AND EFFICIENCY AS OF REGISTERS

the registers, for both hard and soft coal, are shown. With hard coal, it is indicated that whereas 59 per cent of the heat of the fuel was available at the bonnet only 44 per cent finally was delivered at the registers. For soft coal, the corresponding values were 41 per cent available at the bonnet and 31 per cent delivered at the registers.

In designing the heating system for the residence, a register air temperature of 175 F, and a corresponding combustion rate of 7.5 lb coal per square foot of grate per hour in zero weather were used. However, the data taken over a period of two winters indicates that in spite of a furnace efficiency of less than 60 per cent, and a considerable loss between the bonnet and registers, the plant

operated at a 135 deg register temperature and a 3.5 lb combustion rate, as shown by Fig. 7.

The design was based on the assumption that the only heat available to supply the heat loss from the house was the heat delivered at the registers. It is apparent from the data obtained that this was not the case for the house was satisfactorily heated at much lower register air temperatures and combustion rates than those assumed. The difference between the heat appearing at the bonnet

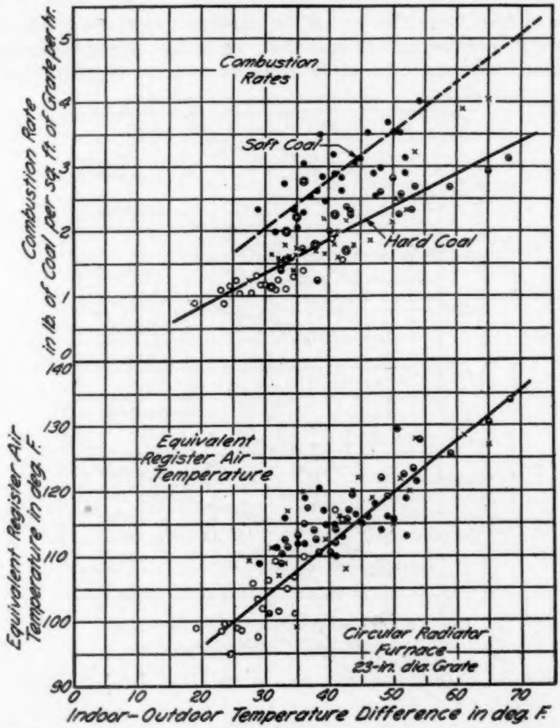


Fig. 7. Register Air Temperature and Combustion Rate Curves for Various Outdoor Temperatures

and that delivered at the registers was not a real loss. The loss from the stacks served to warm the walls and to make up a heat loss that otherwise would have had to have been supplied by the air in the rooms, and the radiation and convection from the leader pipes was available for warming the first story floors. Any heat loss between the bonnet and registers or from the furnace casing and smoke pipe must of necessity remain in the house. The heat may not be utilized to the best advantage, but it is by no means a total loss.

In the same way the heat loss from an inside chimney is available for heating

the house. For an indication of the magnitude of the heat available within the residence from the smokepipe and chimney see the flue gas temperature curves of Fig. 8 and the total heat loss curves of Fig. 9. In the former figure, the flue gas temperature drop in zero weather is (570 — 190) = 380 deg for hard coal, and in the latter figure, the difference in the total heat loss at the furnace and the top of chimney is (20.5 — 7.5) = 13.0 per cent of the heat in fuel.

OVER-ALL EFFICIENCY OF RESIDENCE AND HEATING SYSTEM

The loss of heat from the top of the chimney, when an inside chimney is used, is the only ultimate loss of heat from the house. By subtracting from 100 the percentage losses at the top of the chimney, as shown by Fig. 9, the over-all

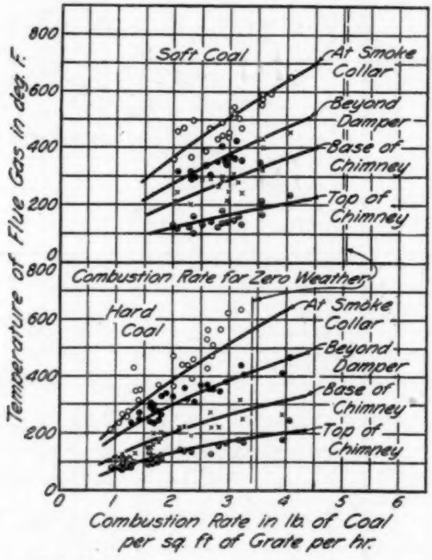


FIG. 8. FLUE-GAS TEMPERATURES FOR HARD AND SOFT COAL

efficiency of the house and heating installation may be determined. This has been done, and the efficiency curves are shown in Fig. 10. When hard coal was fired, the over-all efficiency ranged from 92 to 97 per cent, averaging 95 per cent for average weather. With soft coal, the over-all efficiency averaged 75 per cent.

These over-all efficiencies have also been determined by another method, consisting of an accurate calculation of the heat loss of the building and a comparison of this loss with the heat generated on the grate. A few points based on this method of estimating over-all house efficiencies are shown plotted in Fig. 10 for hard-coal and soft-coal operation. Each point represents the average of several daily tests with each fuel; and the agreement between the points and the

curves indicates that the curves are a fair approximation of the over-all thermal efficiency of the house.

Close agreement between the curves and the over-all house efficiency for any one daily test cannot be expected, since it is quite impossible to make a correct estimate of the exact heating load on any given day. Averages for several days, based on estimated daily heat losses for similar days, are much more reliable values, and were, therefore, used in Fig. 10.

Bearing in mind that the residence was heated to 70 F with high over-all house efficiencies but with low register air temperatures and register efficiencies, it is evident that a large percentage of the heat found its way into the rooms of the house by indirect paths such as through the floors and walls and from

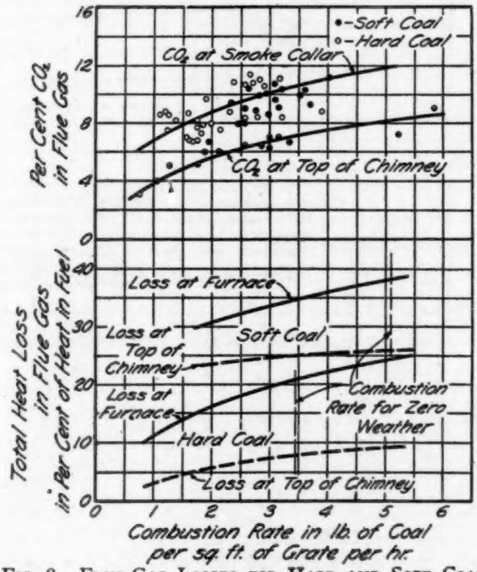


FIG. 9. FLUE-GAS LOSSES FOR HARD AND SOFT COAL

the chimney surfaces. An estimate of this indirect heat is shown in Table 2. With either hard or soft coal, the indirect heat exceeded in amount the heat delivered at the registers.

While the indirect heat exceeded the heat delivered at the registers, its distribution was such that the house was uniformly heated. The average temperature at the breathing level for the first story based on 10 tests, was 70.7 F. For the second story the average was 70.1 F, and for the third story it was 67.0 deg. The average outdoor temperature for the 10 tests was 29.0. The maximum deviations from the average occurred in the S.W. bedroom, where the breathing level temperature was 73.0 F and in the N.W. bedroom where it was 66.4 F. The low temperature in the N.W. bedroom was caused by an unfloored attic space above the room. This condition was later corrected by nailing two layers

of ½-in. quilt insulation on top of the floor joists in the attic. When this was done, the breathing level temperature rose to 69.7 F. Hence, it is evident that the heat delivered at the registers was ample to serve as a control and to produce uniform temperatures throughout the house.

PERFORMANCE ON HARD AND SOFT COAL

The results indicate a considerable difference in plant performance between hard and soft coal. From the general performance curves of Fig. 5 it may be noted that for a given combustion rate, soft coal gave a materially lower efficiency, capacity, and equivalent register air temperature than hard coal. Thus

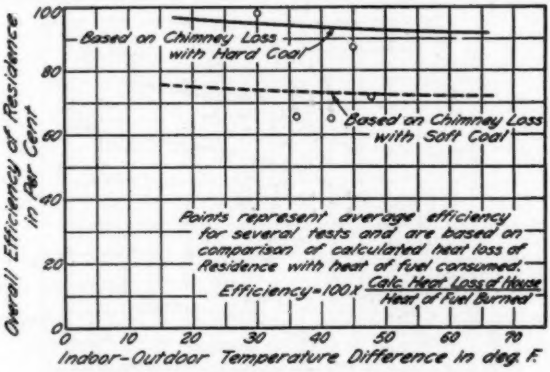


FIG. 10. OVER-ALL EFFICIENCY OF RESIDENCE

soft coal gave an over-all house efficiency of only 75 per cent as compared with hard coal efficiencies of over 90 per cent. An analysis of the data obtained while burning the two fuels indicates the cause of this difference.

The flue gas loss consists of a sensible heat loss and a loss resulting from unburned combustible constituents. Figs. 8 and 9 show the comparison of flue-gas temperatures and heat losses for the two fuels. For each kind of fuel four curves are shown representing the temperature of the flue gases at four points in their passage through smokepipe and chimney. The curves show that at a given combustion rate the flue-gas temperatures were almost the same for the two fuels. Thus at a combustion rate of 3.0 lb the curves show the following temperatures:

	HARD COAL	SOFT COAL
Temperature at furnace, degrees fahrenheit	520	510
Temperature at top of chimney, degrees fahrenheit	175	170

(Temperature at top of chimney actually measured in chimney at roof line.)

Furthermore, it should be noted that at the same combustion rate with hard and soft coal, the CO_2 percentage was the same in both cases. Therefore, the excess air and the weight of flue gas per pound of coal were approximately the same for the two cases. Hence at a given combustion rate, since the weight and temperature of the flue gas was the same for the two fuels, the sensible

heat loss must have been the same. It therefore appears that the difference between the losses with hard coal and soft coal must have been largely due to unburned combustible in the flue gases.

Analyses of the flue gas showed no hydrogen or carbon monoxide for hard coal, but for soft coal they indicated 2 per cent free hydrogen and 1 per cent carbon monoxide. This amount of free hydrogen represented a loss of approximately 1100 Btu per pound of soft coal burned, and the 1 per cent carbon monoxide represented a loss of approximately 500 Btu per pound of coal burned. In addition there was more hydrogen in the soft coal than in the hard coal and hence the loss due to water vapor in the flue gas was greater. The total of these losses amounted to about 15 per cent of the heat value of the fuel.

This 15 per cent loss resulted in higher combustion rates with soft coal for given indoor-outdoor temperature differences than those obtained with hard coal. A contributing factor was the difference in heat value of the two fuels; 11,178 Btu for the soft coal as against 12,618 Btu for the hard coal. The net

Table 2. Percentage of Heat Distribution Through Research Residence

Fuel Used	Hard Coal	Soft Coal
Overall house efficiency, per cent of heat of fuel..........	94	75
Maximum efficiency at registers, per cent of heat of fuel..	44	31
Difference, or per cent of heat of fuel distributed through Residence by indirect paths	50	44

result as shown by Fig. 7 was a 5-lb combustion rate with soft coal for a 70 deg indoor-outdoor difference, as compared with a 3.5 lb combustion rate with hard coal.

It has been indicated that at a given combustion rate the sensible heat loss in the flue gas was approximately the same for both hard and soft coal. But for soft coal the loss due to unburned combustible and increased water vapor together with the lower heat value of the soft coal resulted in a considerable increase in the combustion rate for a given indoor-outdoor difference. This increase was reflected in a higher flue gas temperature, as shown in Fig. 8, where the combustion rate for zero weather is indicated. A higher flue gas temperature represented a greater sensible heat loss, so that for a given indoor-outdoor difference the soft coal also had a greater sensible heat loss in the flue gases than the hard coal. This, however, was not a primary effect, but was brought about directly from chemical losses in the flue gases.

The combined effect of the 15 per cent chemical loss and the increased sensible heat loss in the flue gas accounts for the 20 per cent decrease in over-all house efficiency when burning soft coal. The loss is entirely in the flue gas and could not possibly be available for heating the house.

Conclusions

The practice of comparing heating plants on the basis of boiler or furnace efficiency may be misleading. The efficiency of the heating unit and distributing system is important from the standpoint of delivering heat at desired points, but in a self contained system such as a residence, the vagrant heat by no means

constitutes a dead loss. It raises the heating efficiency of the system as a whole, and generally serves to warm floors and walls where heat is desirable. When the over-all house efficiency is considered, the actual heating unit efficiency becomes less important while combustion efficiency becomes more important.

The over-all effect should be the final criterion for judging the performance of any heating system and the data illustrate that while a study of the performance of the component parts of the system may be valuable in determining the possibility of improvement in the design of such parts, conclusions in regard to the performance of the system as a whole based on a study of the component parts alone may be very misleading.

DISCUSSION

PRESIDENT LEWIS: Professor Kratz has brought up some very interesting points in his paper. I am delighted to see that he has summarized the conclusions which he and his co-workers have reached after a great deal of work on this residence. I am sure that there is going to be some interesting discussion.

JOHN HOWATT: In following Professor Kratz's presentation I noticed that the coal burned is used as a factor in determining the efficiency. In listening to the explanation of the paper it occurred to me that the term should have been the coal fired, not the coal burned. It does not seem possible to obtain an over-all efficiency of 95 per cent on an ordinary house furnace. I know that in an ash test it is not uncommon to find it nearly 30 per cent combustible. If you have 30 per cent combustible in the ash, you cannot have an over-all efficiency of 95 per cent with fuel containing 15 per cent ash. It is possible that Professor Kratz meant the coal thrown in the furnace, not the coal burned when estimating efficiencies.

He shows that a great deal of the heat discharged in the smokestack is recovered in the building by reason of the radiation from the stack. I believe more residences are built with the stack up the outside wall, than there are built as shown in the sketch with the chimney built up through the house. In that case the heat will not be recovered to any appreciable degree.

For a long time I have been interested in trying to learn how much an open fireplace costs. There is an open fireplace illustrated in this plan. I know a great deal of heat goes up an open fireplace. It is a very efficient ventilating device, but it costs money to operate. I wonder if tests have been made in this building equipped as it is with fireplace to determine how much heat is wasted up the fireplace flue.

PROF. A. P. KRATZ: In reference to the first point that Mr. Howatt has brought up, we calculate all of our efficiencies, and all of our performances on the basis of coal burned. We find that we do not have very much combustible loss in the ash pit. Our combustible loss in the ash pit is such that our grate efficiencies run from about 94 per cent to about 97 per cent. The ash is taken out normally once a day, just as it is in house service, and we do not have any carbon in the ash amounting to, as he has said, 30 or 40 per cent. Our total over-all grate efficiency averages 95 per cent, so there would not be very much difference, whether we took it on the basis of coal fired or coal burned.

In reference to his second point, most of the heat regained is not regained from the chimney. There is not such a very large heat loss from the chimney itself. Most of the heat regained is regained from the warm air stacks and

from the smoke pipe in the basement, and from the furnace casing, although the furnace casing does not get very warm.

In reference to the open fireplace, all of our tests have been run with the damper in the fireplace shut, and this is a tight damper. We have tested it with smoke and anemometer readings, and we do not get very much air passing up the chimney. Ultimately, we expect to try leaving it open, and also try it with a grate fire. We have not reached that point in our program yet.

R. V. Frost: I am very glad to see Professor Kratz bring out the point of high over-all efficiency in house heating with anthracite. Assumed low efficiencies have been over-emphasized in house heating, and it is well to have the point stressed to show that we do not have efficiencies of 50 per cent in house heating. It is quite the habit among oil burner and gas boiler people to point out that coal burning is very inefficient; that the average efficiencies run around 50 to 60 per cent. Professor Kratz has shown us that this is not a fact.

In some of my own work in the testing of domestic stokers burning anthracite coal we have found that with these stokers installed in domestic size heating boilers of either steel or cast iron, such as are now on the market, we can easily obtain efficiencies of 85 per cent.

Mr. Howatt's reference to 30 per cent combustible in the ash is a little bit excessive. In our tests we have found that the highest percentage of combustible we have been able to obtain under very careless operating conditions is about 35 per cent, while with normal operation, such as should be obtained in residence heating, the amount of combustible in the ash averaged from 10 to 20 per cent. If you have 10 per cent combustible in the ash, the resultant drop in efficiency is less than 2 per cent and a 30 per cent combustible in the ash means only 7 per cent drop in the efficiency, so even with an excessive drop, this loss is not such a bad characteristic after all. Since the loss in the ash is the heaviest loss in burning anthracite it is apparent that it is easily possible to maintain efficiencies well up to those that Professor Kratz has brought out.

H. R. Linn: We had occasion a few years ago to observe some tests on a warm air furnace. We were aiming to do away with the excess carbon monoxide and unburned hydrogen by introducing secondary air above the fire. I would like to ask Professor Kratz what the difference was in temperature between the floor and ceiling heights in these tests.

Professor Kratz: Our temperature differences between the floor and ceiling throughout the house on the first floor on a zero day averaged from 12 to 15 F. Of course, the temperature difference between floor and ceiling depends on the outdoor temperature. We use zero outside as our basis for comparison. The temperature difference on the first floor averaged approximately 15 F. On the second floor I think it was a little higher; I would say 16 F, and on the third floor maybe as high as 17 to 18 F. This is in common with practically all other types of heating apparatus. You find quite large temperature differences between the floor and ceiling.

In reference to using secondary air above the fire, it has been our experience that combustion rates are so low in warm air furnaces, or any house heating apparatus, that the temperatures above the fire are not maintained at a point high enough to make the utilization of auxiliary air brought in above the fire of any consequence unless you can in some way provide fire arches or use some modification of the down-draft principle. The cracking of the hydrocarbons

occurs at the surface of the fuel bed, and the temperatures are so low that you cannot prevent the cracking of the hydrocarbons even if you do admit auxiliary air above the fire.

PROF. L. M. ARKLEY: I would like to ask Professor Kratz a question, in connection with the distribution of heat in the building. One of the chief criticisms of warm air heating, as I have heard them, was the difficulty of uniform distribution of heat. Also, I would like to ask Professor Kratz if he has any information in regard to the relative efficiency of heating with this system as compared with hot water and steam heating, etc.

PROFESSOR KRATZ: In reference to the distribution of heat around the house, using the breathing line temperatures as a criterion, we found that our maximum differences did not exceed about 3 deg from room to room. Of course, most of the time we had the doors open throughout the house. We ran a certain number of tests with the doors shut between different rooms, and we were not able to observe that there was any particular difference between the distribution with the doors closed and with them open. We found very uniform distribution, therefore, in the whole house.

PROFESSOR ARKLEY: Have you any information as to the efficiency of this system compared with hot water and steam heating?

PROFESSOR KRATZ: Not as to the house itself. We have not run any tests, but there is no reason to believe that it will be any different in this case, at least not materially different.

T. F. MCCOY: I would like to ask Professor Kratz if the air that was heated was taken entirely from out of doors, or if there was any recirculation.

PROFESSOR KRATZ: It was all recirculation. We did not take any of the air from out of doors.

PROF. G. L. LARSON: I assume these tests were taken at a time of the year when you were getting maximum efficiency conditions. Would those hold approximately through the entire season?

PROFESSOR KRATZ: The curves were plotted from results taken over the whole season. We run our tests from the time that we are able to start the fire and maintain a temperature as low as 70 F at the breathing level on through the whole heating season. We find that when the temperature out of doors gets above 50 that we have some tendency to overheat the house, but all of our tests are based on the whole range of outdoor temperatures.

H. GURNEY: The statement is often made with regard to the paramount importance of insulation. Would that apply? Would not one have to measure the heat lost through cellar walls and cellar glass? In most cellars heat is not wanted.

PROFESSOR KRATZ: With a warm air furnace if the system is properly designed the air circulation is free enough through the casing so you do not have a material heat loss in the basement. We have no insulation on the jacket, and our basement temperatures run approximately 60 to 65 F. If the basement is too warm with a warm air furnace system, the way to correct it is not to attempt to apply insulation, but to correct the air circulation system, that is, to reduce the resistance in the air system. I did not mean to imply that insulation is worthless; by no means. Insulation is desirable even in the warm air furnace in order to get the heat up to the rooms where it is desired. But the application of insulation, particularly on a warm air furnace system, will not be re-

flected in savings in the coal pile, or rather, savings in the coal pile will not result from the application of insulation. It merely changes the distribution of heat in the system and not the total loss of heat from the house as a whole.

We have taken humidity readings regularly. Our humidities vary from about 15 per cent to 40 per cent, depending on the outdoor temperature. We have made some attempts at humidifying, using the regular commercial types of humidifiers, but none of them humidify above about 18 per cent in zero weather.

F. W. Johnson: I am somewhat surprised at the 175 F outlet temperature mentioned as having been used in making computations for heat in-put into this particular residence. I have always thought that the furnace method of heating has suffered largely at the hands of its friends. The general public speaks of a *hot air* furnace; whereas, we heating engineers would like to speak of it as a *warm air* furnace. Therein lies the difference between satisfaction and dissatisfaction in a great many installations.

It appears to me that a 175 F outlet temperature in the average residence is excessive. The same amount of heat can be conveyed into a house in a much more healthful manner and distributed through it more efficiently by circulating larger volumes of air at correspondingly lower temperatures. It would appear to me to be obvious that a better distribution of the heated air, in better condition for breathing, with, consequently, more comfortable conditions, can be obtained by such a practice.

I would like to ask Professor Kratz whether this 175 F entering temperature has been accepted and endorsed by the *National Warm Air Heating Association.* If so, it is my opinion that such approval is dictated more by economic considerations than by sound engineering practice, and it can only serve to perpetuate and to some extent strengthen the prejudice which has been aroused in the minds of many against *hot air* furnace heating.

Professor Kratz: In order to answer that I will have to go back slightly into the history of the case. When the investigation was first undertaken, some idea had to be obtained as to what the Association was going to regard as acceptable register temperatures. The Advisory Committee of the Association at that time thought that 175 F was acceptable, so they defined any system operating with register temperatures below 175 F as a warm air system, and any system above 175 F as a hot air system, and went on record as recommending that the warm air system was to be the one used—that is, with register temperatures below 175 F.

When the laboratory tests were run, the leader carrying capacities in Btu per square inch of leader pipe were determined for different register temperatures, and practically all the codes were based on the values at 175 F as the basic temperature for design. As you have seen from the results that we put on the screen, if the 175 F is used as a basis for design, the actual performance of the plant was with register temperatures in the neighborhood of 140, which is, of course, more desirable, as you point out. So the case is automatically taken care of due to the stray heat loss through the house.

Mr. Ainesworth: I think it would be very beneficial and of a great deal of use to consider the use of storage heat from electricity. The *N. E. L. A.* is carrying on rather an extensive investigation along this line. They have had some experience with one home there near Chicago, and an investigation showed that you can heat that home for approximately the same amount with electricity

as you can with gas. That is something to think about. There are several different forms of furnaces that go into this system, and the *N. E. L. A.* is carrying on rather extensive investigations. I think it is something that the Society might take note of, because they are going to get a lot of data out of that which may change some of the formulae for house heating.

C. W. DELAND: I am wondering if you found it more difficult to heat the northwest room with its large amount of wall and glass than the balance of the house.

PROFESSOR KRATZ: The first season that we ran the house, in the northwest bedroom—which was an outlying room—the temperature at the breathing line was about 4 F lower than it was on the average for the rest of the house. We attempted to bring the temperature up in that room by increasing the size of leader pipe, and also by insulating the leader pipe, and we did not succeed. Finally we insulated the room by installing about 1 in. of a quilt insulation on the ceiling joists, and now the temperature of that room is practically the same as the breathing line temperature over the whole house. We did have some difficulty at first and corrected it by insulating the ceiling.

H. L. WALTON: I understood Professor Kratz to account for the lesser observed coal consumption than that predicted as being due to the heat recovered from the loss in the ducts and risers. It occurs to me that there may be a difference in the heat loss of the building from that computed. From the curves there was a certain coal consumption observed, and other curves show that predicted for what I understood to be the same temperature conditions; and that observed was less than that predicted. I understand that the predicted consumption was based on the only heat available for heating the house being that given off at the registers, and that the lower coal consumption resulted from the heat loss in the ducts being effective in heating the house. Now the predicted consumption must have been predicted from computing the heat losses from the building, and the computation of such heat losses is not an exact science. There is usually some error, and might not that account for some of the difference between the two coal consumptions—the predicted and the observed?

PROFESSOR KRATZ: I would say no, because the heat loss calculations from the house itself when compared with the heat in the coal fired, checked very well with our over-all efficiency as calculated from the heat appearing in the flue gas at the temperature existing at the roof line. Therefore, we had that as one check on the accuracy of the heat loss computations. The second check on the accuracy of the heat loss computations was that no room failed to heat by any large percentage of what we had calculated. If we had made a material error in the heat loss calculations, or rather in the assumptions in calculating the heat losses, it would have shown up in some of the individual rooms as well as in the house itself. So I would say to answer your question, that I feel that we have pretty close checks on the validity of our heat loss calculations from the house, both before the system was put in and after the system was installed.

No. 837

AIR CONDITIONING SYSTEM OF A DETROIT OFFICE BUILDING

By Hiram L. Walton[1] and Leslie L. Smith,[2] Detroit, Mich.

MEMBERS

SIXTEEN stories and two basements of the Union Trust Building in Detroit are heated and ventilated throughout by a combined air conditioning and heating system. Some air conditioning of office buildings has been done in the Southwest and provision has been made for it in some recent buildings in other localities, but it is believed that its application to an office building as a combined system and to the extent that occurs in the Union Trust Building would make a description of this installation and considerations affecting its design and operation of interest to the profession.

The Union Trust Building (Fig. 1) is 80 ft wide by 270 ft long on the ground. It is 40 stories in height above grade and has three basements below grade. There are street exposures on the north and south ends and west side and an alley exposure on the east side. The National Bank of Commerce affiliated with the Union Trust Co. occupies the first five floors. The lower banking room is on the first floor and the main banking room is on the second floor and extends in clear height through four stories of the building. Both of these banking rooms are open to the lobby, which is nearly as high as the main banking room and is separated from it for its full width and height (Fig. 2) only by an open grille.

The first basement is occupied by the safety deposit and security vaults and the second basement by the record vaults and filing departments. There is a relatively large working population on this latter floor. The Trust Co.'s offices occupy the seventh to fifteenth floors inclusive and part of the sixth. Provision has been made for the future occupancy of the sixteenth floor.

The space occupied by the Bank and Trust Co. including the basements and sixteenth floor is provided with a combined air conditioning and heating system. By an air conditioning system is meant, a ventilating installation that is capable of supplying a warmed and humidified air in winter and a cooled and dehumidified air in summer accordingly as the comfort of the occupants may

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require. This is further termed a combined heating system when the air is delivered at such temperature as to adequately heat the space supplied. Direct radiation has been installed in the toilets and stairways of the floors in question, but the air conditioning system is depended on for heating the remainder of the space. The space above the sixteenth floor is rental space and is heated by direct radiation of a semi-concealed type.

The location of the safety deposit vaults and the record vaults and filing departments below grade in the first and second basements required a mechanical

FIG. 1. UNION TRUST BUILDING, DETROIT

system of ventilation for these areas for any condition that would be at all satisfactory to the occupants and customers. The first floor banking room has a large number of people on the floor at various times, and this area being only of normal height, 15 ft at the center, mechanical ventilation became a necessity. The main banking room on the second floor is probably of such height and volume that, insofar as ventilation is a necessity, a local air movement within the room might suffice, even though the number of people within this room is considerable during the banking hours. As a convenient means of heating this room and accommodating the heating equipment to the architectural design and treatment, a blast system which in extent and arrangement would correspond to a ventilating installation was found preferable. In the Trust Co.'s offices there is a large working population amounting to a density in some areas of as

much as one person to 50 sq ft of floor area. The Trust Co. had been experiencing an increasing crowded condition in their previous quarters for some time, and their records indicated that the percentage of lost time from illness

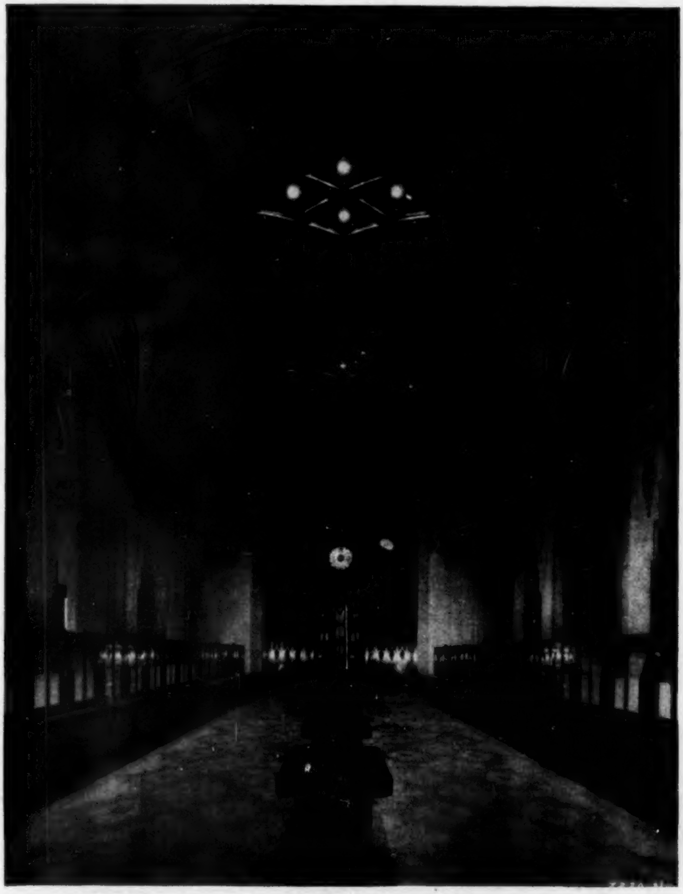

Fig. 2. Main Banking Room with Supply Grills on Columns

of employees increased in proportion to the number of employees that were added in any area when only natural means of ventilation were provided. A comparison with ordinary standards of ventilation further showed that many departments of the Trust Co. could not effectively utilize the space assigned to them if they had to depend on natural and window ventilation. It was evident

that a ventilating installation would be required for both the Bank and Trust Co.'s quarters.

The necessity of a ventilating system having been established, the question

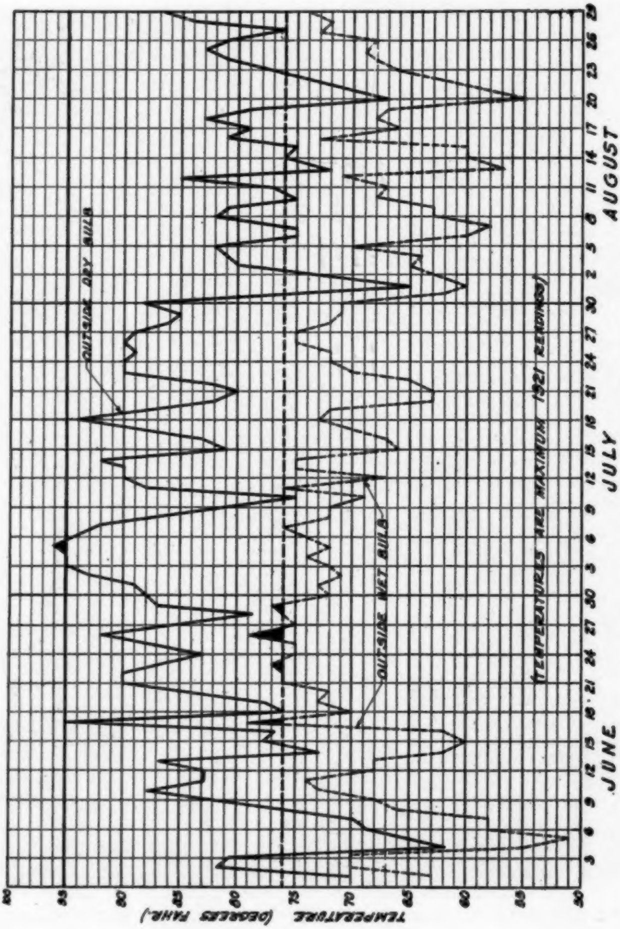

FIG. 3. OUTSIDE TEMPERATURES AT DETROIT

that now required consideration was whether or not this ventilating system should be an air conditioning installation. The additional investment and operating costs could be estimated within such limits as would be required for factors affecting the adoption of air conditioning.

The additional investment was estimated at $130,000.00. The operating costs,

TABLE 1. AIR CONDITIONING OPERATING COSTS

BASIS		
Capacity of refrigeration plant		600 tons
Hours of operation assumed as		500 hours
POWER CONSUMPTION		
Compressors (connected load)	650 hp	
Pumps	90 hp	
Total connected load	740 hp	
Kilowatt Input	555 kw	
500 hours @ \$0.02 per kwh		\$5,550.00
REFRIGERANT		
CO_2 @ 25 lb per ton per season @ \$0.88 per lb		
$600 \times 25 \times .08 =$		\$1,200.00
CONDENSING WATER		
Cost of water: \$0.40 per 1,000 cu ft. Consumption based on 2.9 gal per min per ton with condensing water @ 70 F		
$600 \times 500 \times 60 \times \frac{2.9}{7.5} \times \frac{.40}{1000}$		\$2,790.00
MISCELLANEOUS		60.00
		\$9,600.00

exclusive of maintenance (Table 1), were estimated at \$9,600.00 per year. Allowing annual fixed charges, including maintenance at 11 per cent,[3] the total annual cost of the air conditioning feature over that which straight ventilation would have cost, amounted to \$23,900.00. It will be noted that the operating costs do not include any item for attendance, it being concluded that the air conditioning would not result in any additions to the operating force.

It is not feasible to give a dollars and cents value to the advantages of air conditioning to compare with the additional investment and operating costs. Outside of industrial plants, the widest use of air conditioning has been by theatres and department stores, and primarily for the purpose of attracting trade.

The Bank and Trust Co. were not oblivious to the comfort of their customers, and it was recognized that an air conditioning installation would also have some advertising value for these institutions, but it was believed that comfort for their employees and working conditions that would contribute most to their health and efficiency were considerations that gave the greatest opportunity for return on any investment that might be made. It was futher recognized that there was some element of perfection in having air conditioning, and which

TABLE 2. CAPACITY OF INSTALLATION

System	Lower Bank	Main Bank	Office
Volume of space served (cu ft)	456,500	480,000	1,590,000
No. of people	450	410	2,610
Air supplied (cfm)	69,000	74,000	150,000
No. of conditioning units	1	2	2
Air changes per hour	6.6	9.2	5.7

[3] Interest 6 per cent, amortization 1.5 per cent, taxes 1.5 per cent on full value, insurance and liability 0.5 per cent, maintenance 1.5 per cent, total 11 per cent.

would not be out of keeping with the building and its appointments and the standard of business conducted in it. All features considered, it was concluded

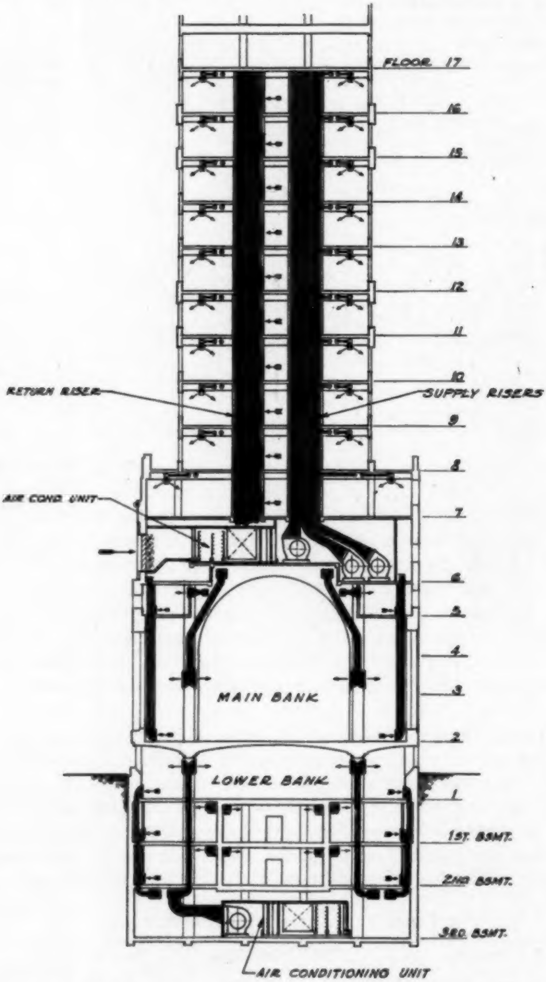

FIG. 4. CROSS SECTION THROUGH AIR CONDITIONED AREAS

that it was well worth while to have the air conditioning at the investment and operating costs that would be incurred.

The conditions to be maintained in the building were established at a tem-

perature of 70 F and a relative humidity of 40 per cent during the heating season, and during the summer a temperature not exceeding 80 F and a relative humidity not exceeding 55 per cent when the outside dry-bulb temperature was 95 F and the outside wet-bulb temperature was 76 F. In arriving at these outside conditions, a chart was constructed typical of summer conditions for this vicinity. This chart is shown in Fig. 3. It will be noticed that the areas above the 95 F dry-bulb line and 76 F wet-bulb line, representing the degree days that the system would not be able to maintain the standard set, are very small. Normally for department stores and theatres in this locality, these limits are set lower (84 F dry-bulb and 73 F wet-bulb) and greater areas are tolerated.

The air conditioning installation for the spaces previously defined consists of three separate systems: First—a system serving the first floor bank, main lobby, and vaults and work spaces occupying the first and second basements;

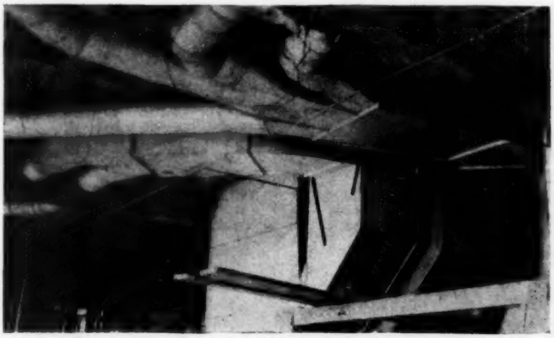

FIG. 5. HEATERS AND DISTRIBUTING DUCTS

second—a system serving the main banking room and office space below the seventh floor; third—a system serving the office floors of the Trust Co. These systems are hereinafter referred to as lower bank, main bank, and office systems respectively. The capacities of these three systems are given in Table 2.

The air conditioning unit for the lower bank system is located in the third basement. The air is supplied (Fig. 4) at the ceiling of the space served and withdrawn from near the floor and thence returns to the conditioning unit. The two conditioning units for the main bank system are located in the machinery room on the sixth floor. This space is formed by the deep trusses carrying the stories above the main banking room and the walls resulting from fireproofing the trusses form the enclosures for the air conditioning units. The admission of air and provisions for recirculation are similar to those for the lower bank system. For the heating of the space served by these two systems, several heating units of the fin-tube type have been installed in the supply ducts. There is a separate heater for each section of the space served whose heating requirements are expected to vary from that of other sections.

The arrangement of these two systems with their supply and recirculating

ducts, the velocities in them and outlet velocities, follow very closely that which may be considered standard practice. It was feared that with the high banking

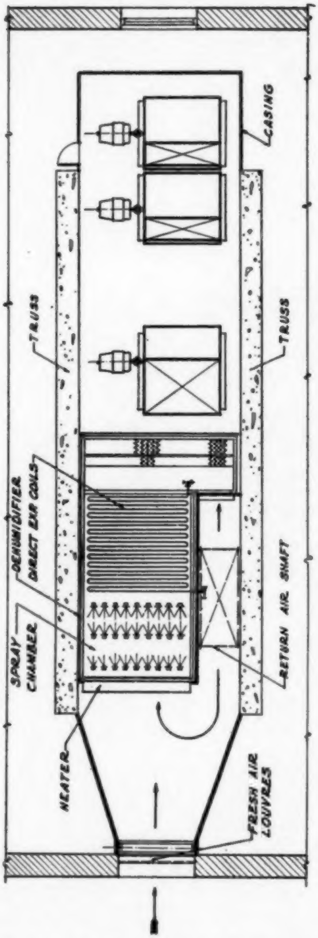

FIG. 6. PLAN OF AIR CONDITIONING UNIT

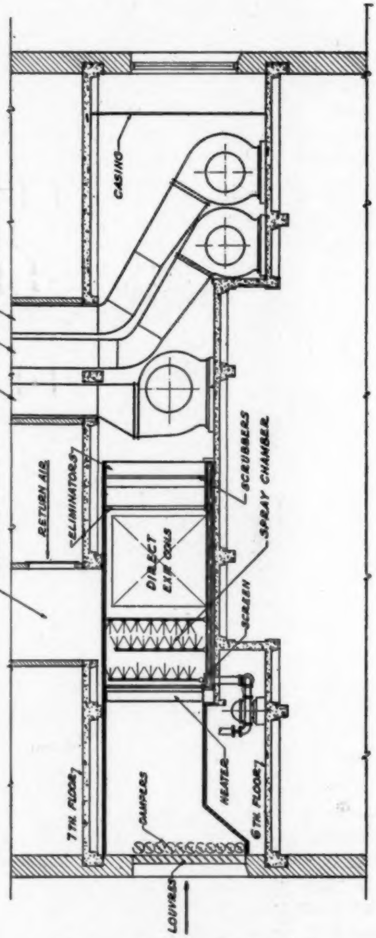

FIG. 7. SECTION THROUGH AIR CONDITIONING UNIT

room open to the lobby, and with the elevators opening to this lobby, there might be objectionable drafts, and that it might be necessary to install a glazed partition between the main bank and lobby. Although the performance has

not been observed under extreme conditions, the indications are that objectionable drafts will not be experienced.

The office system serving the 7th to 16th floors, inclusive, has two air conditioning units installed in the 6th floor machinery room, similarly to those for the main bank system. One of these units serves the North half and the other the South half of these floors. The South unit serves a space having three exposures,—namely, East, South and West. A separate fan, with its supply duct system, has been provided for each exposure, the purpose being to have the system of ducts for any one exposure uninfluenced by variations in the leakage and wind pressure affecting the air supplied by the duct system for any other exposure. It is expected the flexibility of this arrangement will also be of advantage in summer conditions in compensating for the changing sun effect on the sides of the building. A heater (Fig. 5) has been installed on

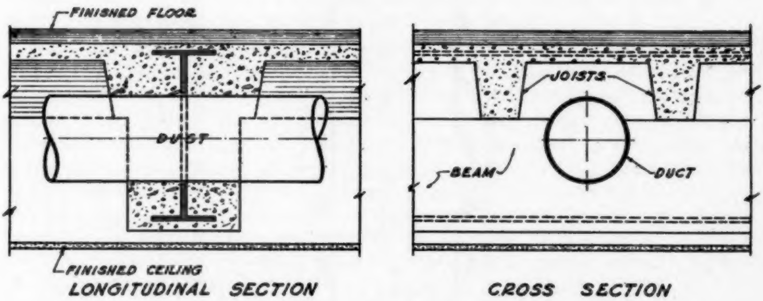

FIG. 8. ARRANGEMENT OF DUCTS AND FLOOR CONSTRUCTION

each floor for each duct system, which can be controlled to give air of the proper temperature to meet the heating requirements.

The air conditioning units (Figs. 6 and 7) for the office system are typical in their arrangement and operation of those for the three systems comprising the complete installation. These units draw outdoor air in through louvres at the sixth floor level. They also draw recirculated air back through the return air shafts connected to return air grilles on the various floors. Both the outdoor air and the return air enter a common chamber ahead of the dehumidifier where they are mixed to maintain a temperature above freezing. Part of the mixed air then passes through blast heaters and through the dehumidifier where it is washed and cleaned and its moisture content definitely fixed. The remainder of the mixed outdoor and return air by-passes the dehumidifier and enters the unit on the fan side. In the summer time the dehumidified air and the by-passed air are then mixed to obtain the required conditions of air delivered to the rooms. The dehumidifying is accomplished by CO_2 direct expansion coils installed as part of the air conditioning unit. During the heating season the process is the same except that the air is humidified.

By running two ducts in place of one for a part of the distance, it was found that the distributing ducts (Fig. 8) on each floor could be kept within a maxi-

mum diameter of 12 in. where they crossed the floor beams. It was further found that structural considerations required these beams to be 22 in. in depth, and that if they were made 24 in. in depth and a joist floor construction used,

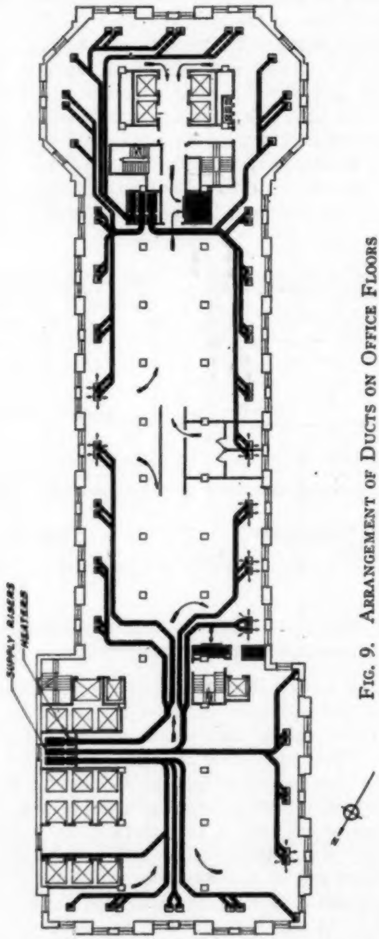

Fig. 9. Arrangement of Ducts on Office Floors

that the 12-in. ducts could run through the web of the beams (Fig. 9), effecting a very considerable saving in the cost of the structure over that which would have been required if the ducts had had to run under the beams. These distributing ducts terminate in a ceiling outlet, there being a double outlet for each bay, such that the usual office partitions can be run through the outlet

if division of the office space at any time requires it. These outlets have a damper for adjustment of the air quantity and a simple diffusing head for directing the air flow along the ceiling. No provision is made for the occupant having any control whatever over the quantity of air supplied to his office or over its temperature.

The office system varies from the bank systems in the provisions for recirculation. There is no system of recirculating ducts other than the recirculating riser to the unit at each end of the building. There are recirculating grills in the office partitions, located in general near the floor so that the air supplied to the office can in part find its way to the corridor and thence to the recirculating risers. Some sacrifice in office privacy has resulted from the installation of these grilles, but in general the owners have felt that they could well tolerate it for the simplification and saving that it effected. In some offices and departments where this loss of privacy was considered objectionable, double grilles have been provided with sound-proofing between them.

With the occupant of an office having no control over the quantity of air supplied or over its temperature, some other provision had to be made for room temperature control. By providing on each floor a sub-duct system for each exposure and providing that system with its own heater, it was expected that, if the system was once adjusted for a proportionate air supply to each office, and if this air was supplied at proper temperature to one office for maintaining the room temperature, it would be of proper temperature for the other offices on the same duct system. On this basis a thermostat has been installed in a typical office for the control of the heater on a duct system. The effect of variation in the air supply due to wind pressure building up a counter pressure on any portion of the space supplied, has been minimized by carrying a relatively high static pressure on the system. Provisions for adjusting for such a condition have been further made by driving the fans, there being a fan for each exposure as above described, with variable speed motors. The occupants are not only prevented from interfering with the operation of the system itself, by adjustment of the quantity and temperature of the air supplied, but they are also prevented from upsetting exposure and leakage conditions by opening windows. All windows on the office floors are provided with keyed locks and are locked closed.

It is felt that the heating of several separate rooms or offices from the same duct system by one heater without means of varying the amount of heat supplied to the various rooms is somewhat experimental even though those rooms may be subject to the same exposure conditions. However, the heating of an office building by a blast system was not untried. Such an installation had previously been completed for the addition to the Detroit Edison Co.'s office building and although of the double duct type, had well demonstrated the practicability and advantages of this means of heating. It was concluded that some greater variations in the average room temperature than are ordinarily considered good temperature regulation might well be tolerated if omission could be accomplished of the usual system of direct radiation. This radiation occupies valuable floor space and subjects those, who must work in proximity to the radiator, to an uncomfortable baking on one side from the radiant heat—hardly consistent with the conditions for an air conditioned area.

There was furthermore a saving effected in omitting the direct radiation

system that was available in offsetting the cost of the ventilating system. 20,-130 sq ft of radiation was omitted which in the form of ordinary wall-hung exposed radiation would have cost $35,000. Presuming that this radiation would have been made the semi-concealed type as was done in the rental space the cost would have been increased for the additional radiation, enclosures, and provisions in the building construction for them, by $16,000.00. The cost of the heater installation replacing the radiation amounted to $26,000.00, making a net saving of $25,000.00.

The work carried on in finishing the building and in moving in has resulted in conditions in the areas served that have made it impractical to obtain more than a preliminary adjustment of the air conditioning installation since its completion. The final adjustments are now being made, and although these have been under way less than a week at this writing, the indications are that difficulty will not be experienced in operating the installation to heat or cool the various rooms within a variation of temperature that will be satisfactory.

The designers of this installation were greatly assisted in perfecting its arrangement and details by the engineers of the B. F. Reynolds Co. and American Carbonic Machinery Co., who made the installation, and by the engineers of other air conditioning companies who gave so freely of their time and experience during the stages of development. The designers are also not unappreciative of the opportunity that was given them by the progressiveness of the building owners.

DISCUSSION

THOMAS CHESTER (WRITTEN): The architects and engineers responsible for this building have displayed praiseworthy initiative and they have taken a radical and progressive step which should be of interest to all members of the Society.

From the information given in the paper it is evident the owners were shown that an air conditioning system was economically feasible and other owners may be expected to follow their example in increasing numbers.

The treatment of the two lower portions of the building can be considered conventional, as many buildings have been similarly equipped. The novel feature is the air conditioning of the 7th to 16th floors inclusive.

It is probable that the degree of success achieved will depend upon the extent of the reduction of infiltration of outside air, which of course is contingent upon the tightness of the windows.

The paper states that effect of wind pressure on the air distributing system has been minimized by carrying a relatively high static pressure on the system. This statement is open to question as the aggregate area of all leakage crevices of windows, doors, elevators, etc. must be so large that no measurable air pressure can be set up within the building. Relatively high static pressure merely connotes smaller and cheaper ducts and higher operating cost.

A 20-mile an hour wind would produce a pressure on the windward side of the building slightly less than 1/5-in. water column and a suction of equal amount on the leeward side. The air movement through the crevices would be at the rate of 20 mph or 1,760 fpm. To offset the wind action the vol-

ume of return air would have to be sufficiently less than the volume of supply air per minute, to permit of the difference supplying the outward leakage of all crevices on the leeward side and also leakages through stairway and elevator doors. In practice, air will be found leaking in through the windward windows and out through the leeward ones, regardless of the interior air distributing system.

The installation of three fan units at each end of the building to permit of alterations in air volumes and temperatures, to compensate for varying wind actions outside the building, is very good and should prove of benefit. This, of course, means diffusing suitably conditioned air with the air leaking in from the outside and this is an entirely different story to any attempt to prevent such inward leakage by carrying a relatively high static pressure on the system. Furthermore the large return air grilles are equivalent to open windows in their effect on air pressure.

According to the paper, air enters an individual office through a ceiling outlet. This entering air will naturally drift towards the corridor, on its way to the recirculating downtake. In cold weather this will probably result in cold drafts from windows being felt by office occupants. It is to be hoped that the entering air is projected towards the windows. Possibly better results could have been obtained by introducing conditioned air through long and narrow grilles in window boxes, this air to be discharged upward in order to mix with the air leaking in due to wind pressure.

M. G. HARBULA: Business men and executives are realizing more and more the importance of air conditioning and the fact that it is not a luxury but a necessity, and above all it can be truly termed as an investment. Of course its application to theatres where more people receive the benefits of air conditioning than in any other type of building has been known as an investment yielding quick returns; in some theatres the investment has paid for itself in less than two seasons of operation. While such instances occur mostly on account of the fact that that particular theatre might have been the first in a neighborhood including other theatres to install a cooling plant, the fact remains that air conditioning is no more a novelty in theatres but an absolute necessity, if an owner wants to have the same patronage in summer that he enjoyed before he installed an air conditioning system.

There is one particular note that struck a responsive chord in me by the authors that is very important; I might refer to that portion of the paper mentioning that the two conditioning units for the main bank system are located in a room the space for which was formed by the trusses carrying the upper stories. This shows collaboration of the architectural requirements of the building and the economical location of air conditioning equipment, which is a very important factor in this work. Another point that cannot be lost sight of by anyone familiar with the design of air conditioning systems is of the same nature and I refer to that portion of the paper where the proper size of ducts was determined by the structural conditions of the floor, as noted in Fig. 8. Too many designers of air conditioning systems forget or neglect the importance of designing air systems with due respect to the building construction, resulting more often than not in increased cost of new buildings or alterations to existing buildings to accommodate air conditioning systems which should not be so if the proper engineering thought is given to such problems. The authors

should be commended on bringing these two points out which to my mind are as important as the correct design and technique of the air conditioning plant itself.

Table 1 shows an estimate of air conditioning operating costs. I have made a thorough study of operating costs of air conditioning plants, it being my good fortune to be so connected with one of my clients that this work was placed in my entire charge. Under power consumption and cost of current the authors use approximately the entire connected load of 650 hp in addition to the pump power to determine the operating cost insofar as consumption of electricity is concerned. While no details of the number of refrigerating compressors are mentioned in the paper, I understand there are three. If this is the case the electrical consumption for a cooling season will no doubt be considerably less than that indicated in the cost; this can be accounted for in several ways. First, the fact that the city water, which I assume is used for condensing purposes, probably does not reach its maximum temperature until August in that locality, consequently each compressor will develop a greater tonnage when water is under 70 F. This in effect would reduce the power consumption for the first four to eight weeks that the refrigerating machines are started in operation in late Spring. Another point that will no doubt show a lower cost so far as electrical consumption is concerned is the fact that the wet-bulb temperature of the outside air does not reach its maximum for the first four to eight weeks of the season that the refrigerating plant is run. This would mean in effect that less power would be used for refrigeration during that period. In other words the cost of operating an air conditioning plant cannot be determined, in my opinion, from the connected load or rated capacity of the refrigerating machinery, which, incidentally are designed for close to a maximum condition of weather, occupancy and temperature of condensing water available. I have found after operating air conditioning plants for several years that the actual cost in a number of different types of plants is far less than any estimates that were available either from manufacturers of equipment or contractors installing same.

A similar analysis of the cost of condensing water brings out the fact that the 2.9 gpm per ton with 70 F condensing water is the amount no doubt determined by manufacturers of standard double pipe condensers as required under the maximum operating conditions set forth in the paper. The average consumption of condensing water over an entire cooling season in a plant that has just ordinary supervision is considerably less than this amount, in fact so much so that it is rather startling to know how little condensing water is actually used in such a plant.

The estimated cost of refrigerant in my opinion is also high, particularly considering the fact that CO_2 plants today are welded propositions throughout. The old days of flanges, bolts and gaskets are things of the past and the modern art welding reigns, with the result that one might safely say the loss of refrigerant in the system is entirely up to the operating personnel for the refrigerant losses would occur in such places as stuffing boxes around piston rods, valve stems, etc., and these can be kept tight by proper attention. In my opinion the refrigerant losses in the system would be considerably less than the amount shown.

I bring all these points out because in my experience I have found that when

anyone contemplates the installation of an air conditioning plant their move is prompted by usually one motive and that is to have cool air; they are not concerned about operating costs and as a matter of fact this is never brought out to them. When, however, they start paying the bills for electricity, water and refrigerant they begin to realize that the "feeding of the horse" is quite a sizable item. Then if such an owner may be in the market again for an air conditioning plant he will think about such things as operating and maintenance cost and if accurate data are available showing such costs he would not be alarmed and would find also that it would not be considered an expense but an investment just as the original cost of the plant no doubt proved itself to be.

The paper in my opinion shows the greater need of air conditioning engineers acting in a consulting capacity; these engineers should be thoroughly familiar with every phase of air conditioning as well as its application to buildings of any type and they should also have a thorough knowledge of building structures and their costs in order to properly and economically adapt an air conditioning system to any type of building. The authors are to be commended for the thoroughness of their work in connection with the Union Trust Building in Detroit.

H. L. Walton: We have since had another period on June 19, in which there were 91 F dry-bulb and a relative humidity of some 60 per cent. Now the question that has been raised by many who have examined the system is as to whether or not the temperature conditions can be controlled reasonably uniformly throughout so extensive an installation, particularly in the office portion. The first seven floors of the building are on one system, as stated before, and it will be observed from the chart on the following page that the dry-bulb temperature for the space served by that system varies from some 73 to 76 F, the wet-bulb temperature from 60 to 64 F, and the effective temperature from the lowest, 66½ F, to the highest, 69.3 F. That was not a variation that was at all uncomfortable to the occupants. It seemed quite satisfactory to them.

In the wintertime we believe the condition is more favorable for the control of the temperatures in the various rooms, since that part of the system on each floor serving one exposure has its own separate heater which is under thermostatic control, and the amount of heat it puts in can be varied as might be required by that one small section of a floor. In the summertime the only opportunity of control is at the dehumidifying unit, and the variation in the amount of air that is supplied to one exposure, is limited, not to one floor, but to all the floors that are on that one riser.

I might say that approximately 70 per cent of the air is being recirculated, and the system is designed to maintain a rather high static in the ducts to get somewhat of a plenum effect to minimize the effect of variations in wind velocities, and the sun effect, or other causes that might change the quantities of air being supplied to rooms on the same duct system.

The observed static pressure is 0.4 in. for one portion of the duct system, and the computed friction loss is 0.24, so that we are consuming some power for the purpose of maintaining that plenum effect. The resulting increase in the cost of power amounts to approximately $600 a year. That is, perhaps, a sizable sum in itself, but it is not a large item in the total operating cost of the system.

This completes the description of the installation.

UNION TRUST BLDG. TEMPERATURES DEG F OBSERVED JUNE 19, 1929

Time	Floor	Location	Temperatures D.B.	W.B.	R.H.	E.T.	
11:00 A.M.	15th	North End	75	63	48	69.3	Units No. 5 & 6
		North Center	75	62	47	68.4	
11:10 A.M.	14th	North End	75	63	51	68.7	
		North Center	72.5	64	63	67.3	
		South Center	75	64	54	69.	
		South End	72	65	69	67.5	
11:25 A.M.	13th	North End	75	62	47	68.4	
		North Center	75	62	47	68.4	
		South Center	75	62	47	68.4	
		South End	75	62	47	68.4	
11:30 A.M.	12th	North End	76	63	48	68.3	
		North Center	76	63	48	69.3	
		South Center	75	62	47	68.4	
		South End	75	62	47	68.4	
12:15 P.M.	11th	North End	76	63	48	69.3	
		North Center	76	61	41	68.65	
		South Center	75	62	47	68.4	
		South End	74	61	47	67.45	
12:30 P.M.	10th	North End	76	63	48	69.3	
		North Center	76	63	48	69.3	
		South Center	74	62	50	67.8	
		South End	74	60	43	67.1	
12:45 P.M.	9th	North End	75	61	44	68.	
		North Center	75	62	47	68.4	
		South Center	73	60	46	66.55	
		South End	74	60	43	67.1	
11:00 P.M.	8th	North End	75	63	51	68.7	
		North Center	75	63	51	68.7	
		South Center	76	63	48	69.3	
		South End	73	61	50	66.9	
1:10 P.M.	7th	North End	75	63	51	68.7	
		North Center	76	63	48	69.3	
		South Center	76	62	44	69.	
		South End	76	60	43	67.1	
1:20 P.M.	6th	Directors' Rm.	70	60	55	64.7	Units No. 1 & 2
		South End	76	60	38	68.3	
1:30 P.M.	5th	North Center	74	63	54	68.3	
		South Center	73	62	53	67.5	
		South End	70	60	55	64.7	
	4th	South End	71	61	56	65.6	
	3rd	South End	72	62	57	66.5	
	2nd	North End	72	62	57	66.5	
		North Center	72.5	62	55	66.9	
		South Center	71	61	56	65.6	
		South End	73	62	53	67.15	
2:00 P.M.	Foyer 1st	North End	74	63	54	68.3	Unit No. 3
		North Center	74	61	47	67.45	
		South Center	74	62	47	67.45	
		South End	74	62	50	67.8	
	1st Bsmt.	Vault Lobby	75	63	51	68.7	
		Deposit Vlt.	73	63	57	67.4	
		Security Vlt.	73	59.5	44	66.5	
		South End	75	63	51	68.7	
	2nd Bsmt.	Record Vlt.	73	61	50	66.9	
		Work Space	75	64	54	69.	

L. A. Harding: I do not think we ought to let this paper go by without discussion. I believe it is a pioneer paper in the field of air conditioning of office buildings. I think Mr. Walton used for the first time in a practical way the term *effective temperature* applied to any discussion that has come up on the floor since we have been given a definition of the term. I would like to ask Mr. Walton whether the $130,000 included the additional space required for the apparatus.

Another item on the fixed charges—I think engineers in general use about 13 to 15 per cent in figuring operating costs. Mr. Walton has, however, indicated the various items that go to make up this charge.

On page 381 I note the number of occupants in the building total up to approximately 2,870, and the total tonnage of the refrigeration machinery is about 600, which is approximately twice the tonnage required for an equivalent occupancy of a theatre. A 3,000-seat theatre takes about a 250 to 300-ton machine, and I suppose this larger size machine was due to the fact that the building has a large exposure compared with its contents as compared with a theatre. I do not know how the number of air changes compares with a theatre. I presume they are less, although the volume of air recirculated seems to be about the same.

James Govan: I would like to point out one feature in connection with this paper that I think engineers cannot afford to ignore. As an architect, I can tell you that today in the modern office building one of the most difficult features that architects have to contend with, and the engineers are going to be confronted with, is that of acoustics. In this paper you have had presented to you the treatment of all the lower floors, which is probably a simple problem, but in your other stories in your office buildings you are going to run up against this thing as engineers: that in the modern skyscraper you have the noisiest conditions that it is possible to have in any kind of building, and today the problem that is being presented to architects and acoustical engineers is how to reduce the annoyance caused by the noises in our cities coming into the buildings, and also penetrating from one part of the building to another.

One of the difficulties you run into as you adopt air conditioning apparatus for high skyscraper buildings is that the minute you connect all of those rooms together with any common system, you are going to run the risk of carrying noises from one room to the other. Just a few months ago this question came up in connection with a building for which I am acting as consulting architect. A New York engineer was also acting as consultant with regard to air conditioning, and when that point was brought to his attention he confessed that the matter had never been given any thought at all. It was the first time this was brought to his attention.

I state that as indicating some of the problems you are going to be up against the minute you apply air conditioning to buildings that have a great number of separate rooms where you are going to connect them together with a system that will carry sound as well as provide air conditioning. The whole problem that architects are up against is to prevent sound from being carried. It is going to require cooperation between acoustical engineers, air conditioning engineers and architects, to see that in improving one condition you do not make the condition that we all suffer from now in the modern skyscraper, worse.

S. R. LEWIS: This is an interesting and pioneering paper. I wish to mention two points on which I have had experience which are at variance with the paper. *First,* I have found it important with cooling systems to have individual control of the various small rooms which have a varying exposure or occupancy or heat-gain. *Second,* I have not found it good practice to install several fans drawing through a washer or filter or tempering heater common to all. No matter what type of fan we have there is difficulty in control and the fans which happen not to be operated must have dampers, which usually are difficult to make tight and which are hard to operate.

DR. E. VERNON HILL: I have not any desire to discuss this paper from the standpoint of design. I leave that to members who are better qualified than I am, but there are two points that are of considerable interest, one particularly, which I want to call your attention to. This is a development in our science which is highly desirable and which should prove of great interest to all of us, the completely air conditioned modern building, but there are two great dangers.

Last week I was in this building in Detroit. I went to see how it was operating. It was not warm enough, however, in Detroit at that time, or it was not cold enough, for me to form any very good opinion as to how it was operating. Earlier in the year I gave a talk to the *Bar County Medical Society* in San Antonio, on ventilation, and I was there three days and spent considerable time in the Milan Building, the only other completely air conditioned building that is in operation that I know of. Well, it was hot when I was down there. The weather was very warm. The condition in the building was ideal. I had no criticism to make, and the owner and the tenants were perfectly satisfied. However, in a warm climate like San Antonio one would naturally suppose that a building that was cooled during the warm season would be very desirable space. On the contrary, the owners of the building have had considerable difficulty in renting space, not because the space is not ideal and the air conditions ideal, but because the people in that vicinity are very skeptical about working in space where the air is conditioned.

There is another new building of very pleasing architecture just completed in San Antonio, and they advertise natural light and natural ventilation very effectively. Now, the point I want to make is this: That it is up to this Society largely to aid in the dissemination of some real facts regarding air conditioning, and I think everyone here should take the position, that the engineer today can bring about and maintain better air indoors than nature provides outdoors. Do not take the position that we are crude imitators of nature; that is not the case. By any scientific test that we are able to use we can prove that the air indoors in these air conditioned buildings is far superior to the air out-of-doors. Do not say that we are trying to imitate nature; we are not; we are going far ahead of nature. Until you gentlemen take that position and preach it, how can you expect the public to believe it? I think that is a sound, logical position to take.

Understand, I do not say it is more healthful to live indoors than it is to live out-of-doors. That is not the case. But so far as the air is concerned we make better air indoors in this present day and age than you ordinarily get in the city out-of-doors. The outdoor life is desirable and healthful on account of the sun effects, exercise, etc. But put over the lesson that the air indoors is easily made and maintained better than the air out-of-doors, and then

you will aid in disabusing the public mind of this thought that the naturally ventilated room is far superior to the mechanically ventilated room.

President Lewis: That is something I think we will have to get the assistance of the medical man on. As I see it, this body of scientific men can furnish any air condition, any degree of elimination of dirt, and any condition of humidity needed for healthful conditions in almost any type of building today. It seems to me the crying need is to have the medical man tell us what, of those various conditions which we can produce, is most healthful for human beings.

S. S. Sanford: A question has been raised regarding drafts from windows under which there is no direct radiation. The chilling effect of the window pane and infiltration of cold air around the window causes a positive downward movement of air in front of windows. This, of course, is lessened by the use of double windows. In most cases the downward movement of air in front of windows produces drafts which are so objectionable that it must be counteracted by a positive upward movement. This upward movement may be produced by heat from a radiator or by introducing air at the bottom of the window with a velocity upward. If it is to be counteracted by introducing air, the temperature of this air must be such that the quantity of heat introduced in this way is sufficient to bring the chilled air from the window up to room temperature, otherwise this chilled air will merely be deflected upward and outward into the room to fall again on the heads of those sitting out in the room in front of the windows.

During the past two heating seasons there has been opportunity to observe conditions in office space having no direct radiation under the windows. In the addition to the Service Building of The Detroit Edison Co., from the 4th to the 10th floors the rooms are heated by introducing air through ceiling diffusers. To counteract window drafts, air is also introduced through window-sill slots with a velocity upward. The temperature of this air is varied from 70 to 90 F, depending on outside conditions. The volume introduced at the windows is much less than through the ceiling diffusers. No double windows are installed. All have wooden sash except for two steel sash windows in one corner. In spite of the precautions which were taken, there have been complaints about window drafts. Apparently the volume and temperature of the air introduced at the windows is not sufficient. Most of the complaints come from two corners of the building, particularly the corner in which the two steel windows are located. This is attributed to the excessive infiltration around these windows combined with the extra heat loss through the steel frames. If it is not possible to increase the temperature of the air from the window slots sufficiently, it may be necessary to install double windows in the two corners or put a small amount of direct radiation under these windows. Possibly an ideal arrangement for an office building would be to introduce air for heating through ceiling diffusers and supplement this with a small amount of direct radiation under each window to eliminate window drafts.

Mr. Walton: I will not attempt to reply to all the questions or comments that have been made, as an expert, but I want to say for my associates and myself that the interest that has been shown is certainly very gratifying, and it would be a pleasure to show any of you the installation should you come to Detroit for that purpose.

With reference to the estimated additional cost for the air conditioning in-

stallation, about which Mr. Harding asked, the $130,000 quoted does not include the space occupied by the refrigerating equipment in the third basement. It is true that that space might otherwise be utilized, but it was believed that its value in such a way as it could be utilized was so small that it was well within the errors of the estimated additional cost of the air conditioning system. Understand, that the additional cost of the air conditioning system is only that necessary to cover the refrigerating part of the installation, and such parts of the dehumidifying units as are required in that connection; and that a ventilating system was to be installed in any event.

With reference to the amount of the fixed charges of 11 per cent—that figure in general seems low, particularly when we consider industrial plant installations of almost any description or power plant work. That is due, I believe, to the item of depreciation, and that a life in many instances of only some 10 years is given, that not being the physical life of the installation by any means, but being a life determined more from the standpoint of obsolescence. A much greater length of life has been allowed here. I don't recall exactly what it is, some 30 years, and instead of using straight line depreciation, the fixed charges have included an item of amortization to take care of depreciation. That would account for the greater part of the difference in the figures as usually taken for fixed charges.

With reference to the tonnage required, it has been much greater than usually required for a theatre in which the population is about the same. I believe that additional tonnage is due entirely to the exposures on the building rather than to the population.

With reference to the discussion by Mr. Chester, particularly on infiltration and plenum effect, I understood that Mr. Chester questioned whether the plenum effect would minimize the effect of wind pressure on the building, and thought that the greater benefit in operating conditions to take care of that condition would come from the separate fans for the various exposures. Whether or not we can carry out the plenum effect on this building to take care of that variation of condition remains to be seen, but just as an abstract principle we might consider several pounds pressure on this duct system. Or, if we consider a compressed air line running through several rooms, and we adjust the openings in that compressed air line to admit a certain quantity of air into each room, I do not believe that we can conceive of any wind pressure on the side of the building that would affect the amount of compressed air being admitted to a room in a measurable amount. Somewhere between this extreme and that of no pressure will lie the advantage that we can take of the plenum effect on this duct system.

I was quite impressed with Mr. Harbula's discussion, particularly with respect to the collaboration by the architects in the design of the building, to provide for such an installation. We are fortunate in that respect in being both architects and engineers, and, of course, we did have the full cooperation of not only the architects but our own structural engineers as well.

I also gathered from the discussion by Mr. Harbula that our operating costs as estimated were high, and that is probably so because we wanted to be on the safe side in representing such costs to the owner, although we tried to have them reasonably close.

In the matter of acoustics and sound transmission, the condition is not at all different from any ventilating system in so far as it pertains to the transmission of sound from one room to another through the same duct. As this building is used as an office building, there has not, so far at least, been observed any objectionable sound transmission from one office to another. Some acoustical treatment has been provided in the grilles that permit the air to pass out of an outside office, a private office, and reach the main corridor, and thence through that corridor to the recirculating shaft. These grilles are double, there being one on each side of the partition, offset from each other and baffled with acoustical felt. That has been done entirely to give privacy to the office. It seems to have been reasonably effective, and conversations carried on within the office cannot be overheard outside.

With reference to Mr. Lewis's comments on the temperature regulation, he expressed a condition that we have considered the most critical in the design of the installation. I can only say that so far the variation in temperatures has not exceeded what can well be tolerated. Apparently it is not as small as some manufacturers of temperature regulating equipment would consider a satisfactory performance for such equipment, but so far it has been entirely satisfactory to the occupants of the building. It is probably much less than results when the offices are heated with direct radiation and the control of the temperature is left to the occupant of the office.

With reference to the three fans drawing their air through the same dehumidifying unit and interference, the fans are of the so-called backward curve blade type. Some manufacturers here can probably better talk on that than I can. We do not have any trouble with the fans interfering or bucking each other, as it is sometimes called.

With reference to Dr. Hill's discussion on the air conditioning of space for office buildings built for rental purposes, I can only say that none of the space that it is expected to rent in this building has any air conditioning equipment; that this is limited entirely to the space that is occupied by the building owners, and they have had a very definite idea of just what is going to be the limitations of such a system. We have their full cooperation and backing in making it a success in their building and the whims and idiosyncrasies of various people, as to what they might like to have in the way of temperature and other conditions, are not recognized. If we can state what the conditions ought to be in a room and maintain them reasonably close to that, any member of the Union Trust Co.'s organization puts up with it, from the president down to the office boy.

With respect to the omission of radiation, and its being a good thing for a man to have to shiver once in a while, that is entirely out of my field. I can't comment on that more than to say that if it is desired to make somebody shiver in this building, we could ordinarily do it.

Regarding Mr. Sanford's information on the experience with their building, the installation was designed in our offices in cooperation with the engineers of the Detroit Edison Co., and it was carried out quite extensively for a portion of their building in a somewhat experimental way.

There are drafts from windows, of course, and I don't think in general that any room can be so equipped with a heating system and ventilating system that

somebody can be placed close to a window and be entirely comfortable there at all times. The amount of draft is going to depend to a considerable extent on the tightness of the windows. I might say that on this building considerable attention was paid to that, and the manufacturers of the sash seem to have gotten an unusually tight window. No tests have been made on it, but the sash manufacturers guaranteed that the infiltration would not exceed ½ cu ft of air per foot of sash perimeter per minute with a 25-mph wind. I believe that it is much tighter than that.

You will probably be interested in one other observation that we have had since the system was in operation, and that is that in one or two of the private offices in which the conditions as measured by our ordinary standards seemed to be just right, the occupants were not satisfied. They felt that it was drafty, as they expressed it. The air movement was not great. We did not measure it, but it was small—I would judge less than 100 fpm—but they were apparently not accustomed to such an air movement and the office was not comfortable to them. I think that is a condition that we are apt to meet in any building which is equipped with such an installation.

No. 838

CAPACITY OF RADIATOR SUPPLY BRANCHES FOR ONE- AND TWO-PIPE SYSTEMS

Co-operative Work by A. S. H. V. E. Laboratory and the U. S. Bureau of Mines Experiment Station, Pittsburgh, Pa.

By F. C. Houghten[1], M. E. O'Connell[2] and Carl Gutberlet[3], Pittsburgh, Pa.

MEMBERS

A STUDY of the capacity of pipe for various parts of a steam heating system has been under way at the Research Laboratory since 1922. A number of reports bearing on various phases of the subject have been published from time to time including a complete and final report[4] on the subject of capacity of up-feed steam heating risers for one- and two-pipe steam heating systems.

This paper deals with the capacity of radiator supply branches for one- and two-pipe systems and gives final results on this phase of the study, which was made by the Research Laboratory, A. S. H. V. E., in cooperation with the Heating and Ventilating Department, Carnegie Institute of Technology.

There are a number of variable factors entering into the capacity of radiator supply branches all of which must be taken into account in making an analysis of the subject. A typical radiator supply branch from a supply riser is shown in Fig. 1. Its capacity is affected by the pipe fittings, *C* to *D* and *E* to *G*, by the capacity of the vertical section of the branch *FG*, by the valve *G*, and by the pitch of the main section of the branch *DE*. The steam carrying capacity of the branch further depends upon whether the condensation from the radiator returns through this branch (one-pipe system) or through a separate return (two-pipe system). In order to have a clear understanding of the operation of such a branch it was necessary to make an independent study of each of these factors.

Fig. 1 is the laboratory set-up on which the study of supply branches to

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[3] Research Engineer, Research Laboratory, A.S.H.&V.E.
[4] Capacity of Up-Feed Steam Heating Risers for One- and Two-Pipe Systems—by F. C. Houghten, and M. E. O'Connell, Transactions American Society of Heating and Ventilating Engineers, 1927, Vol. 33.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Bigwin Inn, Lake-of-Bays, Ontario, June, 1929.

radiator was made. Steam was taken from the 8-in. main, through the 6-in. branch to the 4-in. supply riser *AB*. The radiator supply branch *CH* on which the study was made supplied steam to a cast iron radiator or water cooled condenser *I*, located on a movable platform so that the length and pitch of *DE* could be changed.

When the branch was operated as a one-pipe system, *J* was closed off and the condensate returned through the radiator supply branch and riser and collected and weighed at *L*. When operating as a two-pipe system *J* was opened

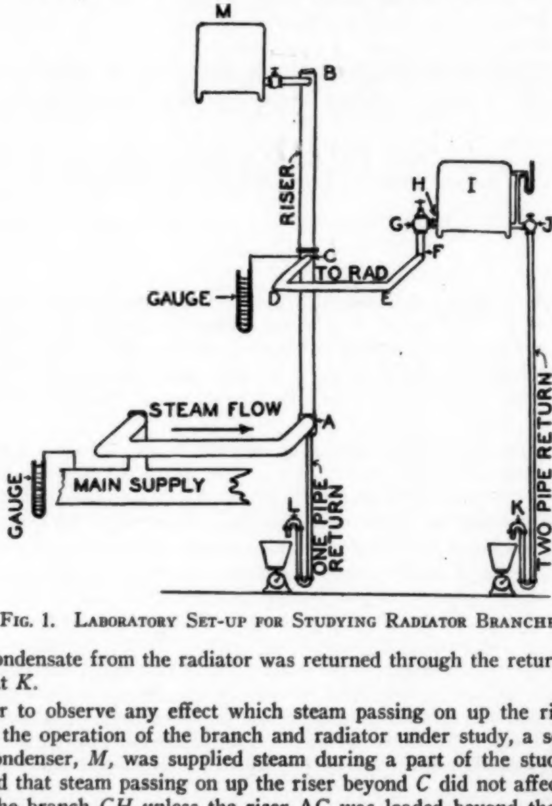

FIG. 1. LABORATORY SET-UP FOR STUDYING RADIATOR BRANCHES

and the condensate from the radiator was returned through the return riser and weighed at *K*.

In order to observe any effect which steam passing on up the riser beyond *C* had on the operation of the branch and radiator under study, a second radiator or condenser, *M*, was supplied steam during a part of the study. It was soon found that steam passing on up the riser beyond *C* did not affect the operation of the branch *CH* unless the riser AC was loaded beyond the maximum capacity recommended for use in an earlier laboratory study[5]. After this fact was established the riser was cut beyond C and capped.

The condensation in the system other than the branch and radiator studied was determined for all operating conditions. When the branch was operating

[5] Capacity of Up-Feed Steam Heating Risers for One- and Two-Pipe Systems, by F. C. Houghten, and M. E. O'Connell, TRANSACTIONS AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, 1927, Vol. 33.

as a one-pipe system the total condensation collected at L was reduced by this amount in order to determine the steam and condensation carried by the branch.

Steam pressures were observed in the main, riser, and radiator and a water column was provided to give the level of any water which might be held up in radiator.

Results

In attacking the problem the relation between steam pressure in the main or the riser and the capacity of the branch as a whole, for any given set of

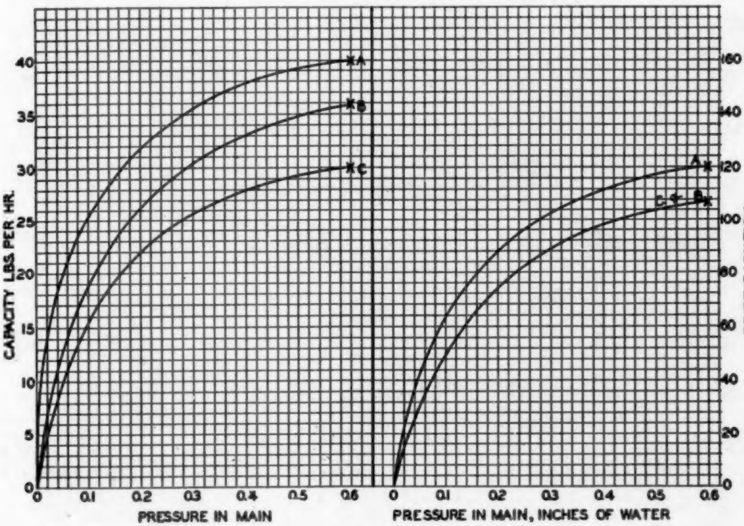

Fig. 2. 1¼-In. One-Pipe Branch. 3½-In. Pitch in 10 Ft.

Curve A, no valve, 1½-in. fittings, 1¼-in. pipe. Curve B, no valve, fittings and pipe, 1¼-in., nominal size. Curve C, valve, fittings and pipe. 1¼-in., nominal size.

Fig. 3. 1¼-In. One-Pipe Branch. 1-In. Pitch in 10 Ft

Curve A, no valve, 1½-in. fittings, 1¼-in. pipe. Curve B, no valve, fittings and pipe, 1¼-in., nominal size. Curve C, valve, fittings and pipe. 1¼-in., nominal size.

conditions, considering the variables mentioned, was determined and the results plotted in the form of a curve such as *A, B,* or *C,* Figs. 2 or 3. These curves are similar to curves showing this relationship in other laboratory publications[6]. In further analysis of the results the maximum capacity, or the highest point x, reached on any particular curve for a given branch was taken as the capacity

[6] a. Capacities of Steam Heating Mains as Affected by Critical Velocities of Steam and Condensate Mixtures—by F. C. Houghten and Louis Ebin, Transactions American Society of Heating and Ventilating Engineers, 1922. Vol. 28, p. 289.

b. Capacities of Steam Heating Risers as Affected by Critical Velocity of Steam and Condensate Mixtures—by F. C. Houghten and Louis Ebin, Transactions American Society of Heating and Ventilating Engineers. 1923. Vol. 29, p. 109.

c. Capacity of Up-feed Steam Heating Risers for One- and Two-pipe Systems—by F. C. Houghten and M. E. O'Connell, Transactions American Society of Heating and Ventilating Engineers. 1927, Vol. 33.

of that branch. Data for such a curve were collected and the curve plotted in order to obtain this maximum point for each of the many variable conditions on which information was desired.

Point *x* on Curve *C*, Fig. 2, is the maximum capacity of a 1¼-in. branch similar to Fig. 1 in which all fittings, pipe, and angle valve were of 1¼-in. nominal size and with a pitch in section *DE* of 3.5 in 10 ft. Point *x* on curve *C*, Fig. 3, gives the maximum capacity of the same branch with a pitch of 1-in. in 10 ft. Curve *C*, Fig. 4, shows the variation in maximum capacity of this branch with pitch.

The valve used in the branch giving the results shown by Curve *C* Figs. 2,

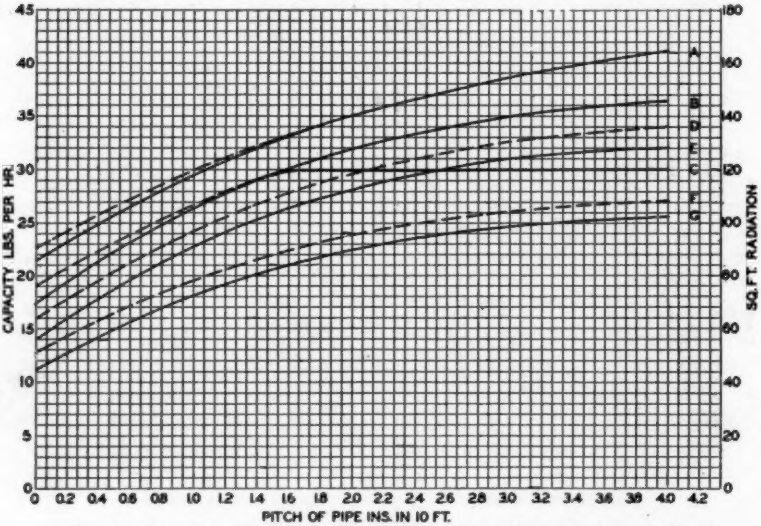

FIG. 4. ONE-PIPE BRANCH. SOLID LINE 10 FT BROKEN LINE 5 FT

Curve A, no valve, 1½-in. fittings, 1¼-in. pipe. Curve B, no valve, fittings, and pipe 1¼-in., nominal size. Curve C, valve, fittings and pipe 1¼-in., nominal size. Curves A, B, and C, maximum capacities with noise. Curves D and E, maximum capacities without noise. Curves F and G 80 per cent of maximum capacity with no noise.

3, and 4 was a standard 1¼-in. angle valve. It was found that this valve and in fact, most valves studied, had a seat opening just 1¼-in. in diameter instead of 1.38 in. in diameter which is the actual internal diameter of nominal 1¼-in. pipe. Another series of data was collected giving curves *B*, Figs. 2, 3, and 4 in which the radiator connection was the same as that giving curves *C*, excepting that the 1¼-in. valve was replaced by a 1¼-in. ell. It was later shown that the same increase in capacity could be had by drilling out the seat of the valve to a diameter equal to that of the same nominal size pipe or by replacing the valve by one of the next larger size.

The data given in Curves *A*, Figs. 2, 3, and 4 were obtained on a similar branch with 1½-in. fittings replacing the 1¼-in. fittings at *CD* and *EG*, Fig. 1. It was further shown that replacing the fittings at either *CD* or *EG* alone with

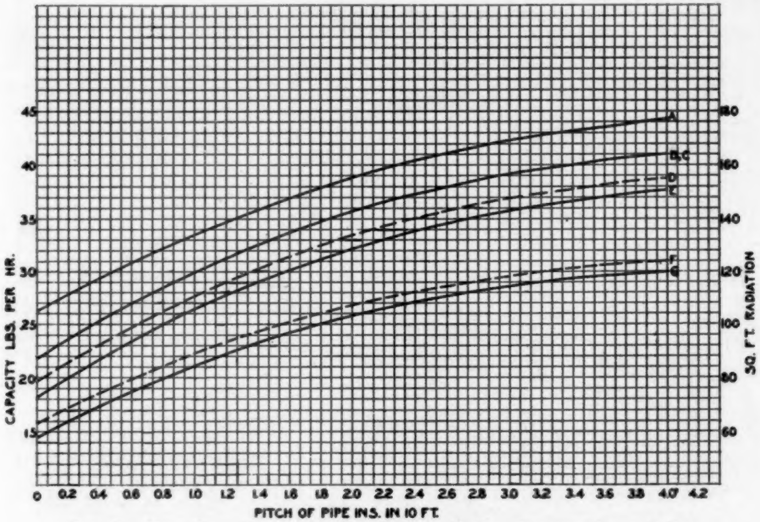

FIG. 5. TWO-PIPE BRANCH. SOLID LINE 10 FT BROKEN LINE 5 FT

Curve A, no valve, 1½-in. fittings, 1¼-in. pipe. Curve B, no valve, fittings and pipe 1¼-in., nominal size. Curve C, valve, fittings, and pipe 1¼-in., nominal size. Curves A, B, and C, maximum capacities with noise. Curves D and E, maximum capacities without noise. Curves F and G 80 per cent of maximum capacities with no noise.

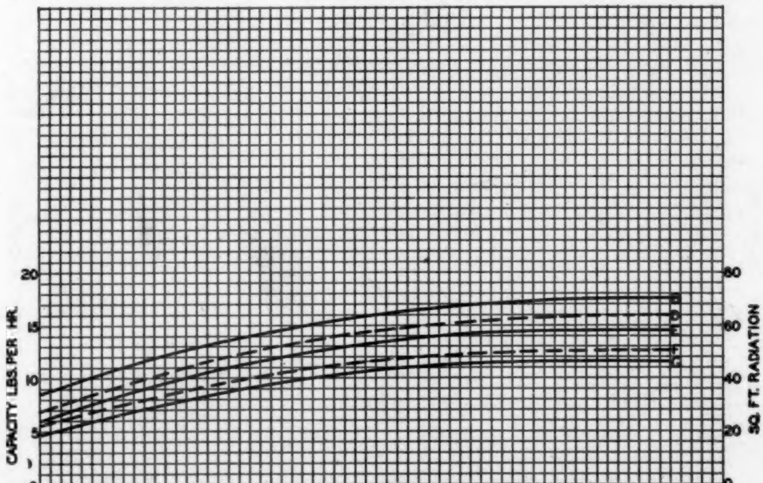

FIG. 6. ONE-PIPE BRANCH. SOLID LINE 10 FT BROKEN LINE 5 FT

Curve B, no valve, fittings and pipe 1-in., nominal size, maximum capacities with noise. Curves D and E, maximum capacities without noise. Curves F and G, 80 per cent of maximum capacities with no noise.

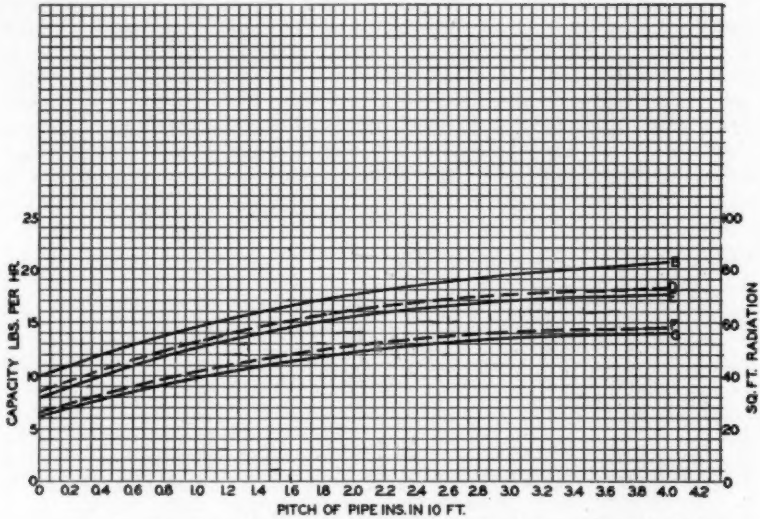

FIG. 7. TWO-PIPE BRANCH. SOLID LINE 10 FT BROKEN LINE 5 FT

Curve B, no valve, fittings and pipe 1-in., nominal size, maximum capacities with noise. Curves D and E, maximum capacities without noise. Curves F and G, 80 per cent of maximum capacities with no noise.

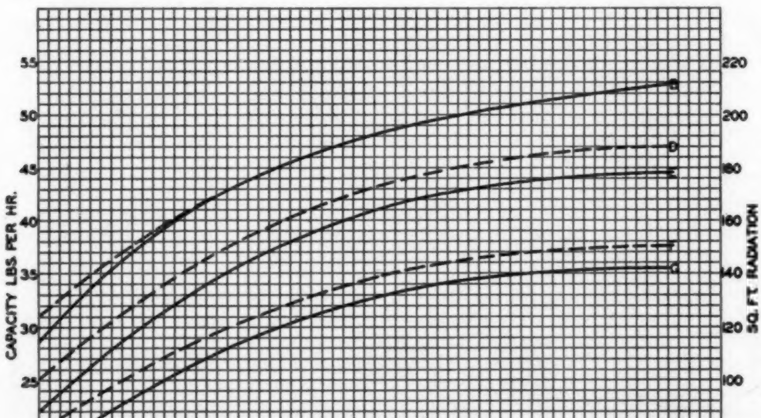

FIG. 8. ONE-PIPE BRANCH. SOLID LINE 10 FT BROKEN LINE 5 FT

Curve B, no valve, fittings and pipe 1½-in., nominal size, maximum capacities with noise. Curves D and E, maximum capacities without noise. Curves F and G, 80 per cent of maximum capacities with no noise.

TABLE 1. CAPACITY OF RADIATOR SUPPLY BRANCHES. ONE-PIPE SYSTEM

Pipe Size	Capacity in Sq Ft of Equivalent Radiation			
	Based on Curves *F* and *G* and pitch of 0.6 in. in 10 ft		Based on GUIDE 1929 Tables 8 and 11, Pitch at least ½ in. in 10 ft	
	5-ft Branch	10-ft Branch	Shorter than 8-ft Branch	Longer than 8-ft Branch
A	*B*	*C*	*D*	*E*
1 in.	31	28	20	....
1¼ in.	68	62	55	20
1½ in.	102	93	81	55
2 in.	189	169	165	81

the larger size gave the same curves as *B*. Hence it is clear that the capacity shown by *C, B,* and *A* in Figs. 2, 3, and 4 are, respectively, the limiting capacities of the horizontal branch alone, the fittings and the valve, of the same nominal size.

Comparison of curves *C* and *B*, Fig. 4 shows that the maximum capacity of the valve was equal to that of the branch with a pitch of 1.58 in. in 10 ft. For greater pitches of the main section of the branch the valve restricted the flow, whereas for smaller pitches the section *DE* of the branch limited the flow.

The data just referred to are all for a 10 ft branch with one-pipe connection (that is for a branch where the condensation from the radiator returns through the supply branch) and they were collected on a set-up where a water-cooled condenser was used to condense the steam, rather than a radiator. With this set-up small noises due to overloaded branches could not be detected easily because of the noise of the water flowing through the condenser and also due to the fact that the small steam condensing space of the condenser did not seem to give the same reverberation to the sound as was experienced with a standard cast iron radiator. The curves *D* and *E*, Fig. 4, give the maximum capacities obtainable with steam condensed in a cast iron column radiator instead of the water-cooled condenser and with the system operating without audible sound. Curves *A, B, C, D,* and *E*, Fig. 5, give data for a two-pipe radiator connection (that is with the condensation returning from the radiator through a separate return branch and riser, similar to that shown in Fig. 4 for the one-pipe branch. For this condition it will be noted that there is no difference between maximum capacity obtainable with a valve of the same nominal diameter as the

TABLE 2. CAPACITY OF RADIATOR SUPPLY BRANCHES. TWO-PIPE SYSTEM

Pipe Size	Capacity in Sq Ft of Equivalent Radiation			
	Based on curves *F* and *G* and pitch of 0.6 in. in 10 ft		Based on GUIDE 1929, Tables 9, 10, 12, 13, 14. Pitch at least ½ in. in 10 ft	
	5-ft Branch	10-ft Branch	Shorter than 8 ft	Longer than 8 ft.
A	*B*	*C*	*D*	*E*
1 in.	36	34	26	....
1¼ in.	79	75	58	26
1½ in.	117	108	95	58
2 in.	209	191	195	95

pipe (Curves *B* and *C*). This is accounted for by the fact that in the two-pipe connection of the type shown in Fig. 1, no condensation returns through the valve. All the steam that condenses beyond the valve returns through the radiator while that condensing below the valve returns through the supply branch and riser. In this system a valve with a small seat will not result in noise unless it is purely a hissing sound due to the velocity of steam through the valve. A small valve, however, will require a higher steam pressure for the same capacity.

The effect of length of a branch on its capacity was investigated by studying branches 5 ft and 10 ft in length. The solid line curves Figs. 2, 3, 4 and 5

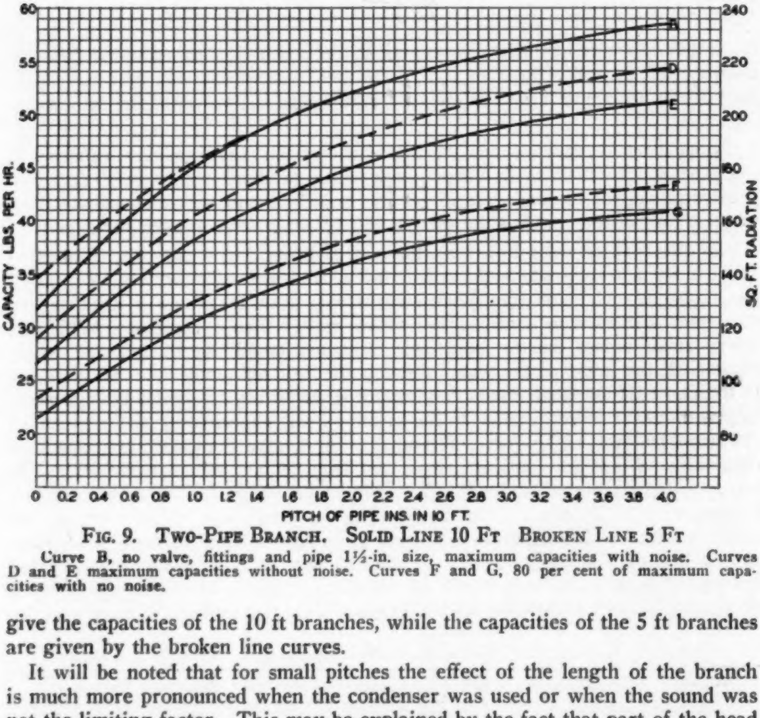

FIG. 9. TWO-PIPE BRANCH. SOLID LINE 10 FT BROKEN LINE 5 FT

Curve B, no valve, fittings and pipe 1½-in. size, maximum capacities with noise. Curves D and E maximum capacities without noise. Curves F and G, 80 per cent of maximum capacities with no noise.

give the capacities of the 10 ft branches, while the capacities of the 5 ft branches are given by the broken line curves.

It will be noted that for small pitches the effect of the length of the branch is much more pronounced when the condenser was used or when the sound was not the limiting factor. This may be explained by the fact that part of the head producing flow in the horizontal pipe may be due to piling up of water at the end from which it is flowing so that the actual diameter of the pipe may be considered to some extent as producing flow of condensate counter to the steam. A perfectly level pipe or in fact one with a small reverse pitch will allow condensate to return, but at a reduced rate. The effect which the diameter of the pipe has on producing flow is a function of the length of the pipe. Hence it will be seen that the possible head varies inversely with the length of the pipe and directly with its diameter. This,. no doubt, accounts for the fact

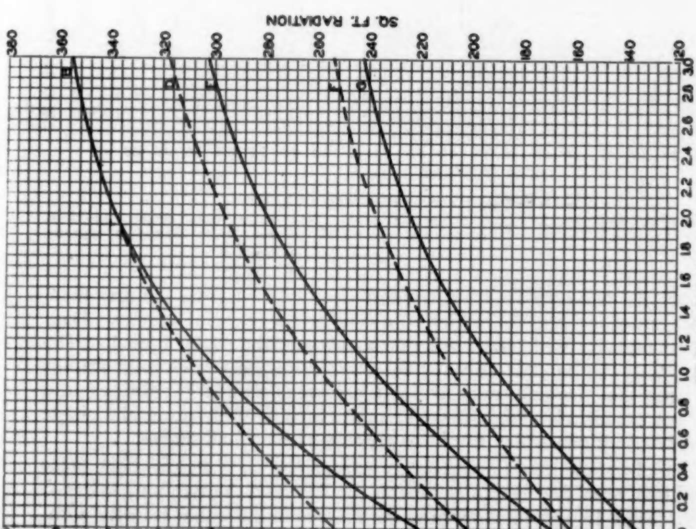

Fig. 10. One-Pipe Branch. Solid Line 10 Ft.
Broken Line 5 Ft.

Curve B, no valve, fittings and pipe, 2-in., nominal size, maximum capacities with noise. Curves D and E, maximum capacities without noise. Curves F and G, 80 per cent of maximum capacities with no noise.

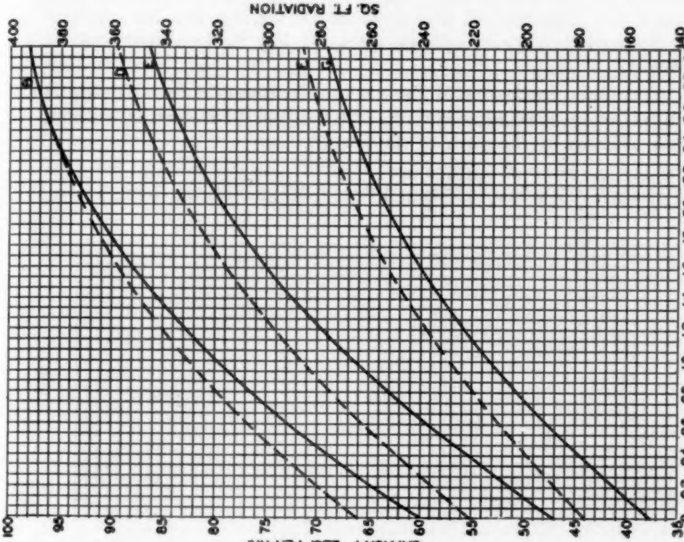

Fig. 11. Two-Pipe Branch. Solid Line 10 Ft.
Broken Line 5 Ft.

Curve B, no valve, fittings and pipe, 2-in., nominal size, maximum capacities with noise. Curves D and E, maximum capacities without noise. Curves F and G, 80 per cent of maximum capacities with no noise.

that the difference in capacity of the 5 and 10 ft length of pipe is more pronounced in the larger branches. It also accounts for the fact that the effect of the length of the branch on its capacity decreases rapidly as the actual pitch of the pipe increases.

When the steam was condensed in the radiators without noise (Curves *D* and *E*) the difference in maximum capacity of the 5 and 10 ft lengths, without noise, was little affected by the pitch of the branch.

Similar data to that given in Figs. 4 and 5 for the 1¼-in. one- and two-pipe branches are given in Figs. 6 and 7, 8 and 9, and 10 and 11, for 1-in., 1½-in.

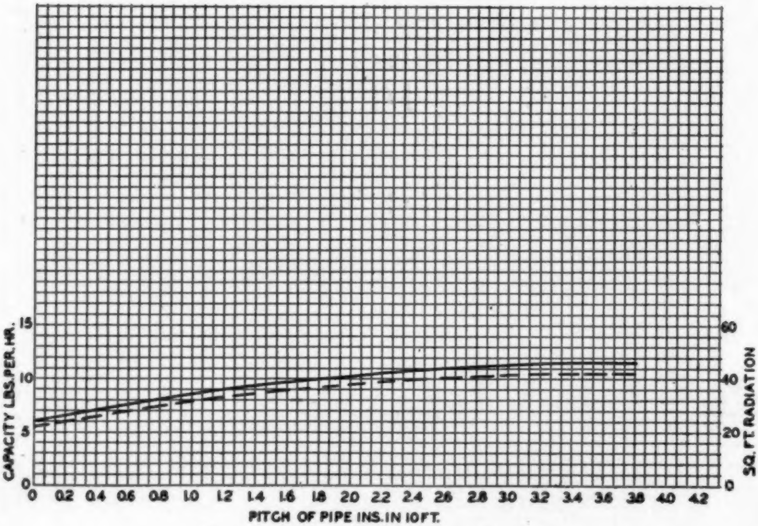

FIG. 12. ONE- AND TWO-PIPE BRANCH

Solid line 5 ft. Two-pipe, no valve, fitting and pipe ¾-in. nominal size. Broken line 5 ft one-pipe, no valve, fitting and pipe ¾-in. nominal size. Both curves are for maximum capacity with noise.

and 2-in., one- and two-pipe branches respectively. Fig. 12 gives the maximum capacity with noise for a ¾-in. branch in a one- and a two-pipe system. Data for maximum capacity with noise are given only for the ¾-in. branch. The curves, Figs. 6 to 11 give data only for the capacity of the branch with fittings of the same nominal size and no valve for the reasons that larger fittings in a branch are not practicable and capacity of valves vary with design and a single figure could not be given which would be representative of all makes.

Figs. 13 and 14 show how the maximum capacity of branches without noise varies with pitch, area, and length of pipe in the one- and the two-pipe systems, respectively. Figs. 15 and 16 show how the maximum allowable velocity without noise varies with pitch of branch and pipe area for 5 ft and 10 ft one- and two-pipe branches.

Practical Application

The purpose of the study was to make available information concerning the flow of steam and condensate in one- and two-pipe radiation branches on which a practical and logical set of pipe sizing tables could be based. This information is given in the curves, *A* to *E*, Figs. 4 to 11 and in Figs. 13 to 16. In making

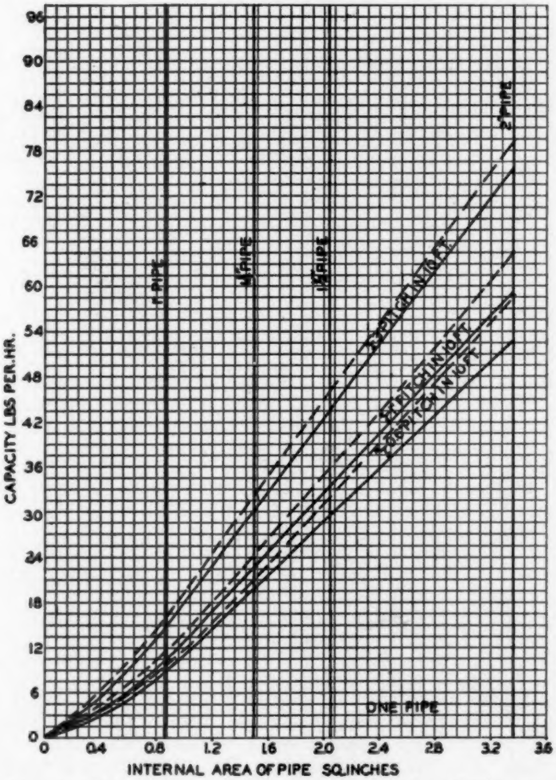

Fig. 13. Variation of Capacity With Pipe Area, Length and Pitch of One-Pipe Branch. No Noise. Solid Line Curves 10 Ft Broken Line Curves 5 Ft

use of these values, however, it should be borne in mind that they are maximum values which could be obtained on commercial pipe under good laboratory conditions. The pipe was well reamed, was new, clean, and straight so there were no sags in the branches.

These conditions will not always be had in practice and some allowance must be made therefore. This allowance will be in the nature of a factor of safety.

A factor of safety cannot be established by laboratory test alone but should also take into account other factors depending on installation and operating conditions, which factors can better be supplied from the practical experience of heating engineers.

Curves *F* and *G*, Figs. 4 to 11 inclusive, give a suggested capacity of branches

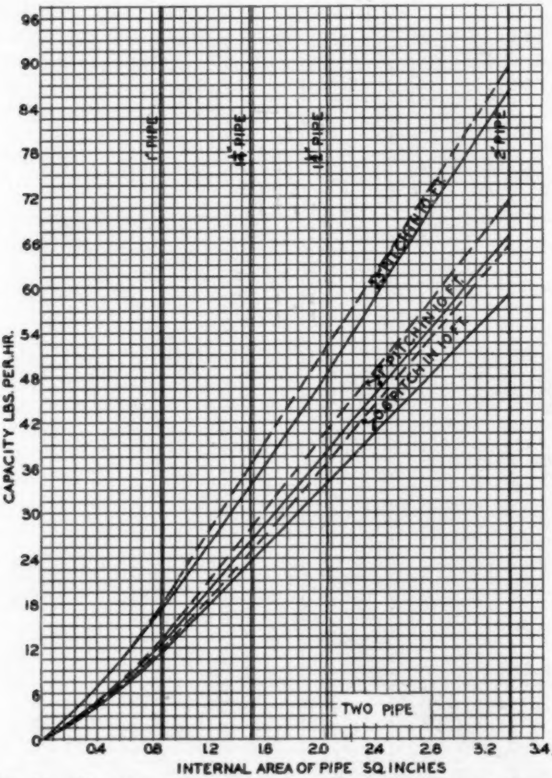

Fig. 14. Variation of Capacity of a Two-Pipe Branch With Pipe Area, Length and Pitch. No Noise. Solid Line Curves 10 Ft Branches. Broken Line Curves 5 Ft Branches

of the different sizes studied based upon a factor of safety of 20 per cent. The capacities given by curves *F* and *G* are 80 per cent of those given by curves *D* and *E*.

It should be pointed out that values given in the curves are maximum capacities or maximum capacities less a factor of safety as specified. No allowance is made for heating-up load and condensation in either bare or covered pipe. Since these factors vary with other conditions they should be considered as factors in calculating the maximum demand for steam.

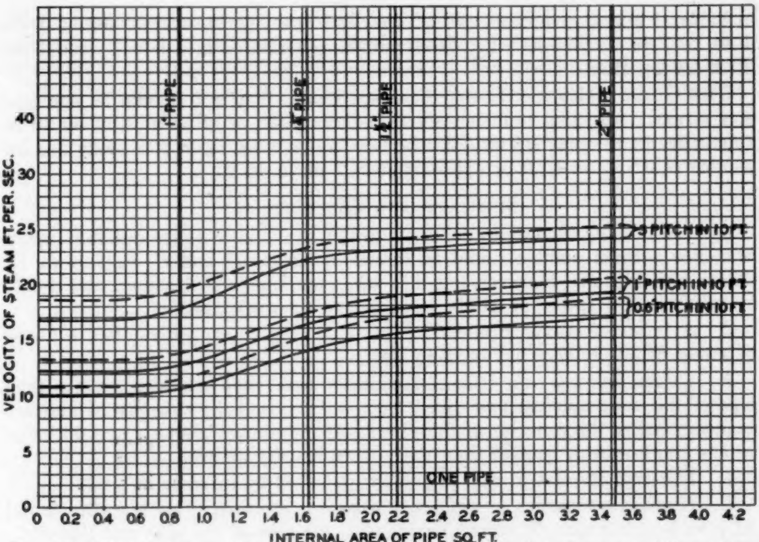

FIG. 15. VARIATION OF VELOCITY OF STEAM IN A ONE-PIPE BRANCH WITH PIPE AREA, LENGTH AND PITCH. NO NOISE. SOLID LINE CURVES 10 FT BRANCHES. BROKEN LINE CURVES 5 FT BRANCHES

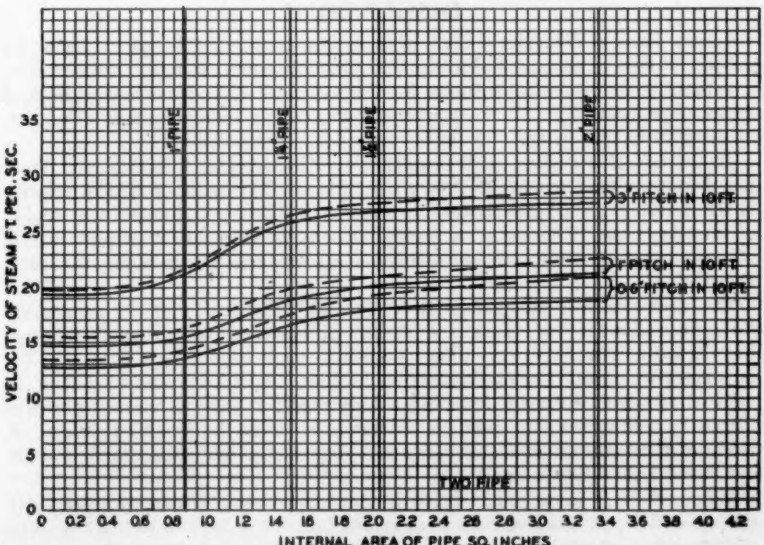

FIG. 16. VARIATION OF VELOCITY OF STEAM IN A TWO-PIPE BRANCH WITH PIPE AREA, LENGTH AND PITCH. NO NOISE. SOLID LINE CURVES 10 FT BROKEN LINE CURVES 5 FT BRANCHES.

Tables 1 and 2 give capacities of one- and two-pipe branches, ranging from 1 in. to 2 in. in size as taken from curves *D* and *E*, Figs. 4 to 12 and from THE GUIDE 1929.

THE GUIDE tables specify that branches longer than 8 ft should be one pipe size larger. Also that branches should have a pitch of at least ½-in. in 10 ft. Capacities in columns *B* and *C* in the two tables based upon this study are for a pitch of 0.6 in. in 10 ft or approximately 1 in. in 16 ft.

SUMMARY AND CONCLUSIONS

1. The investigation gives the capacities of one- and two-pipe radiator supply branches without audible sound as shown in curves *D* and *E*, Figs. 4 to 11. These capacities are, however, only had when the pipe is well reamed and should be taken with a factor of safety for less perfect workmanship.
2. A series of curves *F* and *G*, Figs. 4 to 11 give capacities after a factor of safety of 20 per cent has been applied.
3. Valves found on the market frequently have port openings smaller than pipe of the same nominal pipe size. Such valves limit the capacity of radiator branches below the capacity of the pipe itself for pitches above a certain value which are, however, greater than the pitches usually allowed.
4. The capacity of a branch is greatly increased by increase in pitch.
5. There is a measurable difference between the capacity of branches 5 and 10 ft in length which difference varies with the size of the pipe and the pitch. The larger the pipe the greater the difference in capacity with length. This difference is also more pronounced for small pitches.

DISCUSSION

LOUIS EBIN (WRITTEN): All data as presented in this paper upon the subject of flow of steam, as well as in the previous paper, seem to be based upon a definite point of critical velocity for every size of pipe, for one-pipe as well as two-pipe systems. (To this writer who has done a limited amount of experimenting upon this subject, this is not an exactly true condition).

In the one-pipe system, with counterflow of condensate, it is true that there is a definite critical velocity for every size of pipe. This point of critical velocity is the dividing point between smooth, even, stable flow and a permanent region of unstable flow. This point of critical velocity is independent of pipe length, of steam pressures, etc. In my opinion, it probably is a true point of critical velocity.

But when there is steam flow in a pipe with the return of the condensate through a separate line, there is a distinct difference of occurring phenomena. Fig. A shows a series of curves as obtained through experiments by the writer on some 1-in. pipes. Curve *A* indicates the results of a 1-in. pipe 10 ft long, in vertical position. Curve *B* indicates a 1-in. pipe 5 ft long in vertical position. Curve *D* indicates a 1-in. pipe 10 ft long in an exact horizontal position. A study of these curves indicates some very interesting information not brought out in the Laboratory's reports.

Where there is some condensation in the supply riser, returning contrary to the steam flow, as in all vertical lines and in horizontal lines of positive pitches, the curves indicate a flattening out at one stage (Curves *A*, *B* and *C*). But whereas in the one-pipe system, this point indicated a definite dividing line

between the stable and unstable condition, in the two-pipe system, I would term this point merely *a point of temporary instability.* Below this condition there is smooth continuous flow, and above it there is smooth continuous flow.

Now this point of temporary instability or critical velocity, if you wish, varies

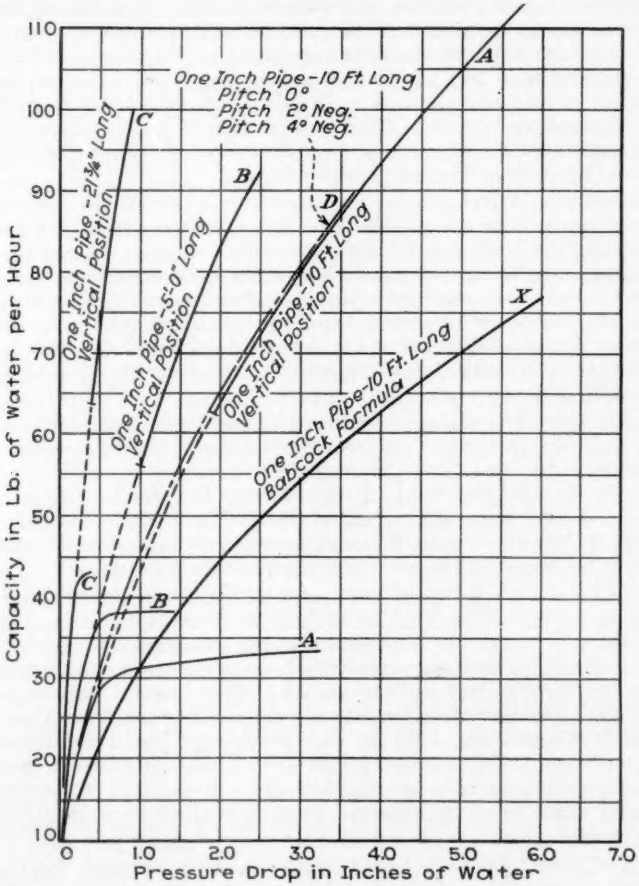

FIG. A. RESULTS OF TESTS ON 1-IN. PIPES. TWO-PIPE SYSTEMS

very definitely with the length of the pipe, and from my curves it seems altogether likely that for very short pieces of pipe it disappears entirely, and we have only one continuous smooth curve. Also it is altogether likely that for very long pieces of pipe, say 30 ft or longer, the flat portion drops considerably in value. In view of this, can the Research Laboratory say, that their tables of capacities, based on 10 ft lengths of pipes with a certain factor of safety, will

still provide a satisfactory factor of safety for risers 30 ft or larger? I believe this is an interesting point.

This flattening out portion of the curve is due to an entrance effect caused by the down-coming condensate and the up-going steam. The quantity of down-coming condensate is a function of the length of pipe. Now suppose the pipe is covered, thereby decreasing the amount of condensate in the steam supply riser. Is it not likely that the flattening portion of the curve will be raised, and as the amount of heat lost through the covering becomes less and less, the flat portion of the curve will tend to be entirely eliminated and the curve become continuous and smooth? Might it be possible to increase the capacity of two-pipe risers merely by covering them efficiently? I believe this should be an interesting point to the heating engineer.

When the pipe is pitched in the negative direction, so that the condensate in the steam supply riser flows in the same direction as the steam, we find that the flattening out portion of the curve has entirely disappeared, and the curve is continuous and smooth. Furthermore for the same length pipe the results obtained are the same whether the pitch is 0 deg, 2 deg or 10 deg negatively, also the only difference between this curve and the curve for the vertical pipe (Curve *D* vs Curve *A*) lies in the flattened out portion of the curve, for the vertical pipe. Above and below this portion, the curves are identical for the same length of pipe.

Another point of interest is a comparison of the experimental results obtained for the two-pipe system and the theoretical results obtained by means of the Babcock formula. Invariably within the limits tested by this writer, whether it be ¾-in., 1-in., 1¼-in. or 1½-in. pipe, for the same pressure drop, the experimental results obtained are considerably greater than those calculated by the Babcock formula. Yet the Babcock formula has for years been the standard method of the heating engineer for figuring his steam flow tables.

The sum and substance of all this is that the writer for one, has more than a vague notion that steam capacities are possible far in excess of anything used or advocated at present. It is possible to obtain consistent and smooth flow of steam in a pipe, using much smaller piping than we are now accustomed to. Witness the large 4-story building known to the writer, in which a vacuum system was converted from an upfeed to a downfeed system. The risers, 2½-in. at the bottom and 1 in. at the top, were not changed. A 1-in. horizontal arm was run from the main to the risers. Yet this system with the risers apparently installed upside down is functioning perfectly with no noise or interferences furnishing ample steam throughout. There is food for thought and investigation in this.

W. H. CARRIER: Some time ago, I think it was last year, about this time, the *Guide Committee*, Perry West the Chairman, and Mr. Haynes and myself, discussed the question of the conflicting data on return mains, especially with vacuum systems. I think we had various accepted data that gave ratings anywhere from two or three to one. Is that right, Mr. Haynes? Greater than that, he says. There was no unanimity of opinion at all. I think a part of that was due to the fact that without automatic regulation, which is becoming more and more universal, you could use very small sizes, continuous operation. There is no question but under such conditions you could use very small sizes,

but in practical operation you are turning these radiators on and off by hand or automatic control where sudden changes of demand are required, and where great difficulty may be found in the tendency for a vacuum to be formed, especially in unit heaters or in ordinary radiators, due to the fact that the vacuum formed would be greater than anything produced in the vacuum system and tend to pull the condensate back into the lines, unless the lines were sufficiently large and so arranged that air in the return system and in the receiver would return back counterflow to the flow of condensate to equalize the pressure.

This last test that Mr. Houghten showed was to be a test for this purpose. Now you get an entirely different condition in continuous operation than you do from intermittent operation—sudden stoppage of the steam and suddenly turning the steam on again. It is a very critical condition and causes water hammer annoyance, water logging of your surfaces, etc., if the pipes are not sufficiently large. Of course, one remedy would be to have a vacuum breaker to prevent excessive vacuum. The other remedy, the more logical one if it could be worked out as the intent of these tests as far as our committee was concerned who recommended them, was to see what size of pipe would be required under different conditions for preventing this condition of excessive vacuum, preventing the condensate being drawn back through the vacuum valves or traps, or whatever was used in the vacuum system, into the radiator.

I want to emphasize while Mr. Houghten is here the importance of this particular thing. The continuous flow proposition is a very simple thing, but the intermittent flow is an entirely different proposition and has to be treated in an entirely different manner, both as to size of pipe and method of experimental operation. I think if this thing can be solved it will be a great service to the heating industry. At present I do not think it is solved. In fact, I know it is not from the results that occur; the fact that water does collect. I do not know that it can be solved easily. We may find as a result of our experiments in the Laboratory that we have to take some further step as a practical solution than simply giving a sufficiently large pipe size, but if we can do it with a sufficiently large pipe size that is probably the simplest solution.

There is one other point in the radiators tested. I presume these were under continuous operating conditions. In heating up a radiator you have an increased demand, especially with a vacuum system, a heating-up factor requiring condensate considerably in excess of the normal condensate, and this should be included in the factor of safety in some logical manner, depending upon the type of radiation used, and upon its heating-up capacity.

H. R. Linn: In Fig. 1, I am wondering if the condensing power of radiator *M* does not have a good deal of influence on that branch connection. And turning to Table 2, I wonder if we cannot answer the disparagement between the values of *B, C* and *D,* due to the temperature of the air in which that radiator stands.

Professor Willard ran some tests on both of these conditions about five or six years ago, and I think he found entirely different results from these.

F. E. Giesecke: I regard this a very admirable paper and believe the work, so far as it concerns one-pipe systems, has been completed, but so far as it concerns two-pipe systems, additional work should be done. In this connection, I believe it would be well to adopt a definition of a two-pipe system. In my

work, I consider as two-pipe systems only those in which the steam and the condensate flow in the same direction. According to this classification, the pipe *D-E*, as Mr. Houghten has already pointed out, is a one-pipe system and not a two-pipe system, because the condensate is returning against the flow of the steam. This probably explains why increasing the pitch of the pipe will increase its capacity. If there were steam without condensate flowing in the pipe, a slight change in pitch would not affect the capacity of the pipe. If the upper radiator connection were used and the pipe pitched downward toward the radiator, the conditions would be very different and the capacity of the pipe much larger.

I believe it would be well to expand this investigation so as to include higher steam pressures. It seems to me that the pressures used so far have been so low that for a pipe 10 ft long the capacity of the pipe is practically determined by the friction head in the pipe.

R. V. FROST: Some years ago Professor O'Bannon of the University of Kentucky, carried on some experimental work on the flow of air and water in return lines, but the work was dropped for some reason. I do not know why, but I think it was very unfortunate for the Society that this was done. I think it would be very wise for the Research Laboratory now to start cooperative work with the University of Kentucky to complete that series of tests. It is about half completed, and I believe Professor O'Bannon has the apparatus all set up ready to proceed. At least it was still there when we were at the meeting at Lexington. I wish the Laboratory could continue that work.

PERRY WEST: Referring to what Mr. Carrier has said in regard to the purpose of the test on return pipe sizes, one of the principal purposes of these tests is to determine the effect of the length of the run, both the horizontal and the vertical run, on the size of return pipes. This is a matter upon which there is a great difference of opinion; as to whether the length of run has anything like the same influence upon the capacity of return pipes that it does upon the capacity of supply pipes.

C. V. HAYNES: I think we ought to carry on that work. The *Heating and Piping Contractors National Association* in conjunction with the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS are now cooperating very closely. Mr. Hart, you know, is the chairman of that committee, and they are asking us to proceed with that work and carry it to an end. Mr. West and others know all about it, and Mr. Harding, and I think Mr. Houghten can carry it through successfully.

L. A. HARDING: It is the intention of the Laboratory, I believe, to continue the very excellent tests that have been run on dry pipe sizes by Professor O'Bannon. He ran tests on 1 in. only, and there is a need for continuing this work. I do not see any very great need for experimental work on vacuum return pipe sizes. The real difficulty is due to the use of copper type radiation in connection with a hot blast system under thermostatic control. I do not think we will correct it by increased pipe sizes. I believe a mechanical device, as suggested by Mr. Carrier (vacuum breaker), however, may be the solution.

F. C. HOUGHTEN: In his written discussion, Mr. Ebin suggests the possibility of considerably increasing the capacity of pipe in a two-pipe system. If the pipe is covered so that less steam will condense in it, it will have a slightly

greater capacity because the capacity of pipe in a two-pipe system is a little greater than a pipe of the same size in a one-pipe system where all the condensation returns through the supply piping. However, since the decrease in condensation in a pipe in a two-pipe system, due to covering it, is a small percentage of the water returned through a similar pipe in a one-pipe system it would not seem possible to greatly increase the capacity due to covering.

As pointed out by Mr. Ebin the capacity of a two-pipe riser can be greatly

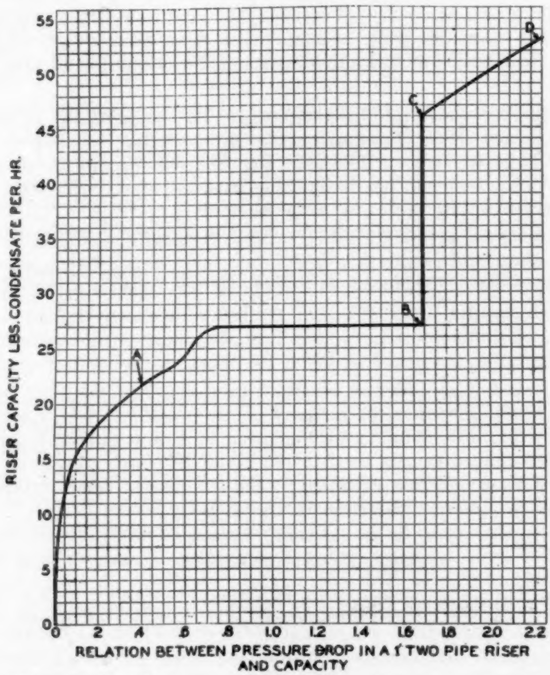

Fig. B. Variation in Capacity of 1-in. Two-Pipe Riser with Increasing Pressure Drops

increased if you want to force the condensation in the riser upward, so that it will return through the return piping.

Fig. B gives the variation in capacity of a 1-in. two-pipe riser with increasing pressure drops as determined by the Laboratory. The section of the curve zero to *A* represents capacities for perfectly smooth operation. The condensate returns through the riser with no interference from the up-flowing steam. As point *A* is passed, interference between up-flowing steam and down-flowing condensate begins as can be observed in a glass pipe. This interference is described in an earlier Laboratory report and increases as you go from *A* to *B* on the curve. Throughout this part of the curve all the condensation in the pipe

does, however, return downward through the riser and no noticeable sound results. If the pressure drop is increased beyond *B* the condensation in the riser is suddenly swept upward and the capacity suddenly increases to *C* and then continues to increase to *D*, and indefinitely beyond, as the pressure drop is increased. You will get a similar but somewhat different curve for a horizontal pipe.

There have been two points of view held by different authorities in regard to allowable capacity of pipe in a steam heating system. Mr. Donnelly and others, I believe, are firmly of the opinion that pipe in a steam heating system should be sized below the point of disturbance *A* in order to get uniform and proportional distribution of steam to all parts of the system for all rates of steam consumption.

Other authorities including those responsible for accepting the values now in THE GUIDE tables, while recognizing that proportional distribution will not be theoretically perfect from *A* to *B*, feel that distribution will be so nearly perfect that a system designed on the limiting capacity at *A* will operate entirely satisfactorily.

A system designed to operate above *B* or in the region *CD* will, however, be so out of balance most of the time and so apt to be noisy that it would not be feasible to base the design of steam heating system on this higher capacity.

Mr. Carrier and Mr. West discussed the last slide shown dealing with a different phase of this subject on which the Laboratory is now working. The questions which they raise will be given consideration in continuing the study.

Mr. Linn asked if the amount of steam carried on up the riser beyond the branch studied did not affect the capacity of the branch. A 4-in. riser was used in all of this work. We could not get radiation enough at *B*, Fig. 1, even though a water-cooled condenser was used to reach the ultimate capacity of the riser. However, for a considerable variation in the quantity of steam carried beyond *C*, there was no apparent variation in the capacity of the branch. I imagine that when you come close to the ultimate capacity of *A-B*, so that considerable water is mixed with the steam, you will get water carried out into the branch causing trouble.

Mr. Linn questioned the location of the radiation referred to in the tables. The capacity in square feet of radiation given in the table and charts does not refer to the size of the radiator used, but the steam or heat in Btu that was transmitted. A square foot as used in the paper means 240 Btu per hour, or in other words, pounds of steam actually handled. Whereas the radiator was surrounded by air at various temperatures, it was picked of such a size, regardless of its surface, that it condensed the number of pounds of steam given.

No. 839

PIPE SIZES FOR HOT WATER HEATING SYSTEMS

The results of cooperative research between the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the Texas Engineering Experiment Station

By F. E. GIESECKE[1] (*Member*) AND ELMER G. SMITH[2] (*Non-Member*), COLLEGE STATION, TEXAS

INTRODUCTION

THE object of this investigation is to secure data which may serve as a basis for tables and graphs from which pipe sizes for hot water heating systems may be easily determined.

The heating systems studied were of the gravity flow type with under-foot main and of such size as to be suitable for residences of ordinary size.

Three varieties were studied, namely:

(a) A system with single main, as shown in Fig. 1.

(b) A system with double main and direct return, as shown in Fig. 2.

(c) A system with double main and reversed return, as shown in Fig. 3.

In each case the heater was located about 3 ft 6 in. below the mains, the water was heated by means of steam and all pipes were bare. The temperature of the water was measured by means of ordinary mercury thermometers passing through rubber stoppers so that the thermometer bulb was immersed in the water.

The outstanding result of this investigation is the discovery that, for the type of system studied, the pressure heads produced by the individual radiators with their connecting pipe lines may be so large when compared with the pressure head produced by the heater with its main flow and return risers that they may interfere seriously with the proper circulation of the water in a gravity

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flow system of the type used in this investigation, if the pipe system has not been accurately designed and built.

These disturbing effects of the radiator pressure heads are particularly active during the heating-up periods of the operation of the heating systems. For example, in the system shown in Fig. 4, the pressure head produced by the heater tends to produce flow through the system in the directions shown by the straight line arrows. The pressure head produced by Radiator 1 tends to produce flow through the system in the directions shown by the wavy line arrows. The hot water reaches Radiator 1 before it reaches Radiator 2, and the pressure head of Radiator 1 becomes active as soon as its flow riser is filled with warm water. In the section between the heater and Radiator 1, the two pressure heads supplement each other, but in the sections between Radiator

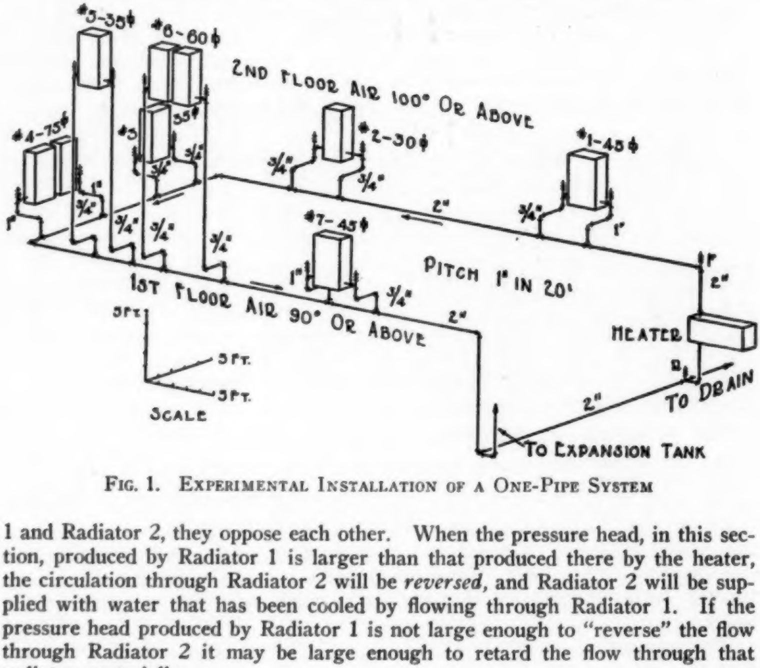


FIG. 1. EXPERIMENTAL INSTALLATION OF A ONE-PIPE SYSTEM

1 and Radiator 2, they oppose each other. When the pressure head, in this section, produced by Radiator 1 is larger than that produced there by the heater, the circulation through Radiator 2 will be *reversed,* and Radiator 2 will be supplied with water that has been cooled by flowing through Radiator 1. If the pressure head produced by Radiator 1 is not large enough to "reverse" the flow through Radiator 2 it may be large enough to retard the flow through that radiator materially.

The opposing pressure heads produced by intermediate radiators in two-pipe direct return systems may be largely responsible for the sluggish flow through the radiator farthest from the heater.

The *reversal* of flow through Radiators 5, 6, and 7 in the two-pipe direct return system is clearly shown in Fig. 21. However, this *reversal* was also found in Radiators 6 and 7 in the two-pipe reversed return system (the so-called, non-short-circuiting system) as shown in Fig. 15.

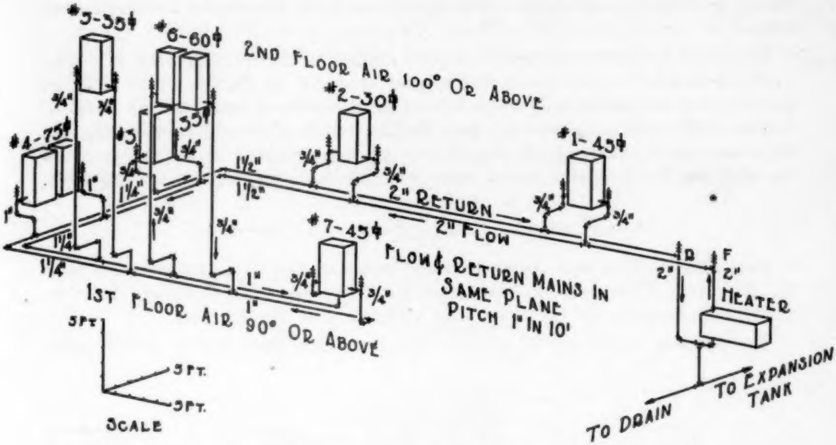


FIG. 2. EXPERIMENTAL INSTALLATION OF A TWO-PIPE DIRECT-RETURN SYSTEM

To prevent the *reversal* of flow in all parts of the heating system, the installation must be designed and executed so that at all points of every circuit the available pressure head will be of the direction necessary to produce the flow desired by the designing engineer.

The above explanation may be clearer if the heater and radiators are replaced by circulating pumps as shown in Fig. 5, or by electric generators, as

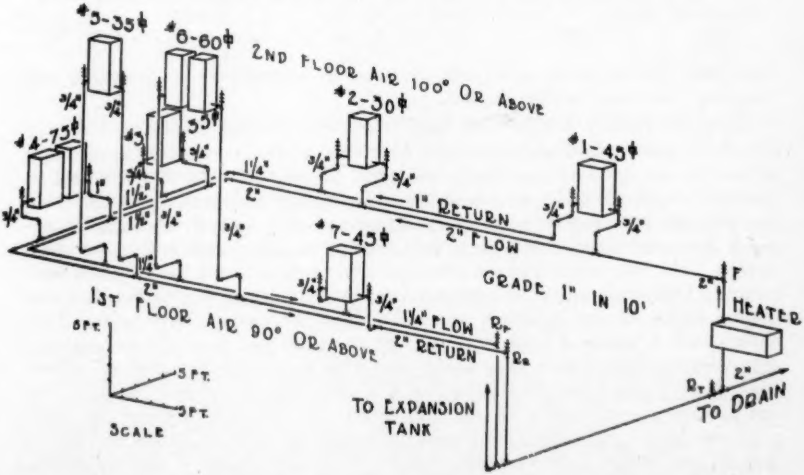


FIG. 3. EXPERIMENTAL INSTALLATION OF A TWO-PIPE REVERSED-RETURN SYSTEM

shown in Fig. 6, and, in that case, electric currents substituted for streams of water.

To design heating systems of the types studied in this investigation it is evidently desirable to have the pressure head produced by the heater as large as possible and the pressure heads produced by the radiators as small as possible. These results can be attained in part by placing the flow and return mains as high as possible above the heater and as near as possible to the radiators, and by attaching the flow and return risers to the bottom tappings of the radiators.

EXPERIMENTAL INSTALLATION AND OPERATION

This investigation was planned and is being conducted in co-operation with H. M. Hart, Chairman of Committee on Pipe Sizes for Heating Systems, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

The purpose of the investigation is to secure data upon which simple tables

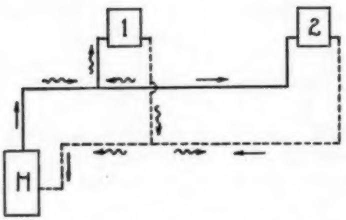


FIG. 4. DIAGRAM ILLUSTRATING HOW THE PRESSURE HEAD GENERATED BY ONE RADIATOR MAY CAUSE A REVERSAL OF FLOW IN SOME OTHER RADIATOR

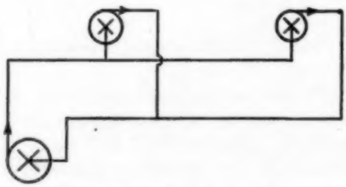


FIG. 5. CIRCUIT ANALOGOUS TO THAT OF FIG. 4, THE HEATER AND RADIATORS BEING REPLACED BY CENTRIFUGAL PUMPS

and rules can be based which may be used by steamfitters in designing and planning hot-water heating systems.

Since the correct design of a large hot-water heating system is not very simple, it was decided to experiment first with a small system such as might be used in the ordinary one-family residence. It was assumed that it would be possible to prepare tables by which the ordinary one-pipe heating system could be designed, but since a properly operating two-pipe system possesses advantages over the one-pipe system, it was decided to experiment with a two-pipe system. Mr. Hart preferred the two-pipe direct return to the two-pipe reversed return. One objection to the two-pipe reversed return was that the flow and return mains in the basement generally slope in opposite directions. This makes such a system a little more difficult to install and causes it to present a less attractive appearance than is the case with a direct return system. However, since many engineers believe that a system with a reversed return is less apt to *short circuit* than a system with direct return, it was believed that such a system would adapt itself more easily to design by means of tables or rules of thumb than would a direct return system. For this reason it was decided to begin the experiment with a reversed return system. It was felt that the objec-

tion to having the two mains sloping in opposite directions could be easily overcome by pitching both mains downward and connecting the far end of the flow main to the return, as shown in Fig. 3, so as to permit complete draining of the system when desired.

To arrive at some basis for an empirical table of pipe sizes, it was assumed that the distance from the center of the heater to the center of the flow and return mains would be about 3 ft 9 in. and that the temperature drop through the system would be about 30 deg, and that the total equivalent length of circuit of any radiator would be about 150 ft. For these conditions the total available pressure head[3] would be about 500 mil-inches of water column and the permissible friction head in the pipe lines, about 3½ mil-inches per foot.

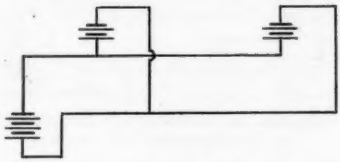


FIG. 6. ELECTRICAL CIRCUIT ANALOGOUS TO THAT OF FIG. 4, THE HEATER AND RADIATORS BEING REPLACED BY BATTERIES

For this condition the following table of pipe sizes[3] would apply:

Pipe Size	Btu
¾	4,200
1	7,500
1¼	16,000
1½	24,000
2	44,000
2½	75,000

Using this table, the pipe sizes shown for the mains in Fig. 3 were selected.

No pipe smaller than ¾ in. was used and, consequently, all risers were made ¾ in. or larger.

In all tests, a heat exchanger was used as a heater and the water heated by means of steam having a pressure of 25 lb or less. The installations were made in a building in which the air temperature ranged from 90 F to 110 F during the tests.

The system shown in Fig. 3, as first installed, had four radiators, Nos. 1, 2, 3, and 4, on the first floor, and two radiators, Nos. 5 and 6, on the second floor. Radiator No. 7, as shown in the figure, was not at first installed.

When this system was put into operation Radiators 1 to 4 operated correctly, but Radiators 5 and 6 did not. During every test run the flow of water was reversed either through Radiator No. 5 or through Radiator No. 6. Fig. 15 illustrates the records secured during one of these unsatisfactory tests. The flow

[3] These values were taken from the diagram published in the 1929 GUIDE and in the Design of Gravity Circulation Water Heating Systems by F. E. Giesecke.

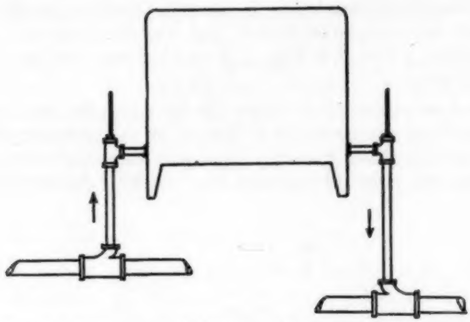


FIG. 7. METHOD OF INSTALLING LONG SWEEP TEES IN THE REVERSED-RETURN SYSTEM

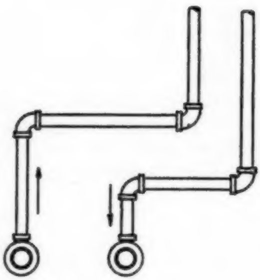
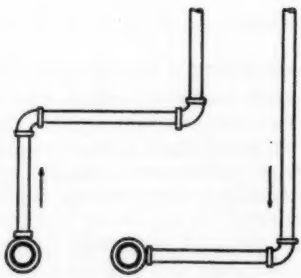


FIG. 8. RISER CONNECTIONS TO MAINS AS ORIGINALLY MADE

FIG. 9. RISER CONNECTIONS TO MAINS AS MADE IN AN ATTEMPT TO PREVENT REVERSED RADIATOR FLOW



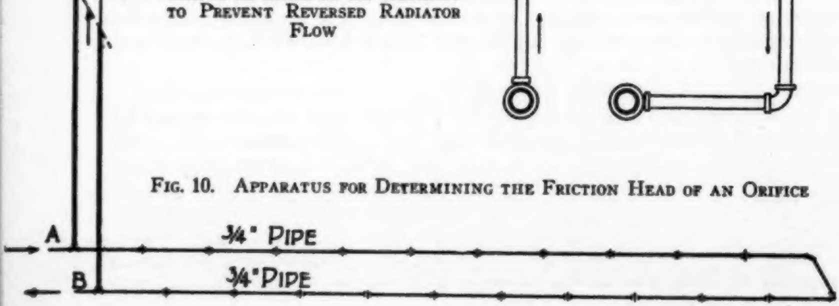


FIG. 10. APPARATUS FOR DETERMINING THE FRICTION HEAD OF AN ORIFICE

was never in the same direction through both radiators at the same time; sometimes it would be direct through No. 5 and reversed through No. 6, and at other times, reversed through No. 5, and direct through No. 6.

In order to remedy this condition, the following changes were made: (1) The ordinary tees connecting the risers of Radiators 5 and 6 to the mains were replaced by long sweep tees, as shown in Fig. 7, in such a way that the friction in the circuit was materially increased whenever the direction of the flow was reversed.

This change had practically no effect; the flow continued to reverse in one of the two radiators, in spite of the additional friction caused by this reversal in the long sweep tees.

(2) The riser connections to the main were changed from that shown in Fig. 8 to those in Fig. 9. This change also had no apparent effect.

While the changes enumerated above were not effective, it seemed at times

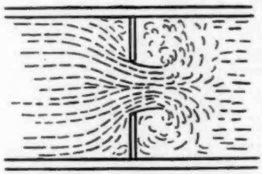


Fig. 11.—Flow of Water Through a Submerged Orifice

as if they were helping considerably, indicating that the dirction of flow was unstable, and might be reversed by comparatively minor influences.

(3) Up to this time the flow and return risers had been connected to the lower tappings of the radiators; as it seemed that the direction of the flow was unstable and that a small difference in pressure head or in friction head might serve to reverse the flow or to prevent reversal, the connections of the flow risers were changed from the lower tappings to the upper tappings of the radiators so that if a reversal of flow through any one radiator occurred, the hot water would have to enter that radiator through the lower tapping and the cooler water leave through the upper tapping. This change was also ineffective, and, in fact, it made conditions worse, as it resulted in a reversal of flow not only in Radiator No. 6, but also in Radiator No. 3.

At this time a careful study was made of the pressure heads producing flow and the friction heads resisting flow in the several parts of the system, and especially in comparison with the conditions that exist in an electrical installation in which the heater and the several radiators are replaced by batteries which furnish the pressure heads, and the pipe lines are replaced by wires which furnish friction heads and it was decided to introduce sufficient friction in each radiator connection to balance the pressure head produced by that particular radiator and also the pressure head available at that radiator of the total pressure head produced by the heater—in other words, to design the circuit for

every radiator so that the friction head in that circuit is exactly equal to the pressure head for that circuit.

Since this could not be done without using pipes smaller than ¾ in. for the radiator connections, it was decided to use one or more circular orifices, installed in unions in the radiator connections. For this purpose copper sheets,

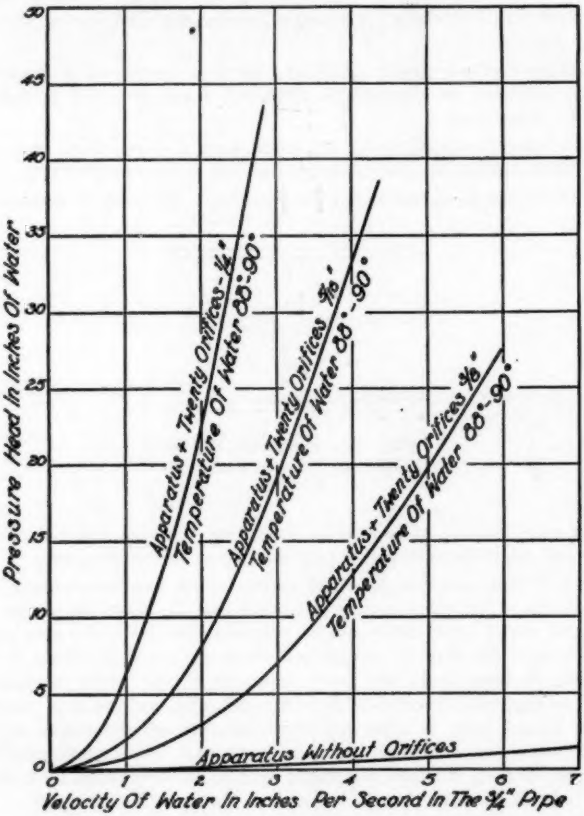


FIG. 12. WARM WATER FRICTION HEADS OF THE APPARATUS DEPICTED IN FIG. 10.

having a thickness of about 0.025 in., with openings of respectively ¼ in., 5/16 in., and ⅜ in. were secured and the friction heads produced by these orifices in ¾ in. pipes were determined.

To determine the friction head caused by a circular orifice in a thin metal plate placed in a union in a ¾ in. pipe, the apparatus shown in Fig. 10 was constructed in such a way that there were 20 unions separated from each

other by 6 in. nipples, the pipe line doubling back on itself so that the two glass manometer tubes were near each other to facilitate the measurement of the difference in elevation, H, of the surface of the water in the two tubes. This difference in elevation, H, is the head lost in the path from A to B. The larger

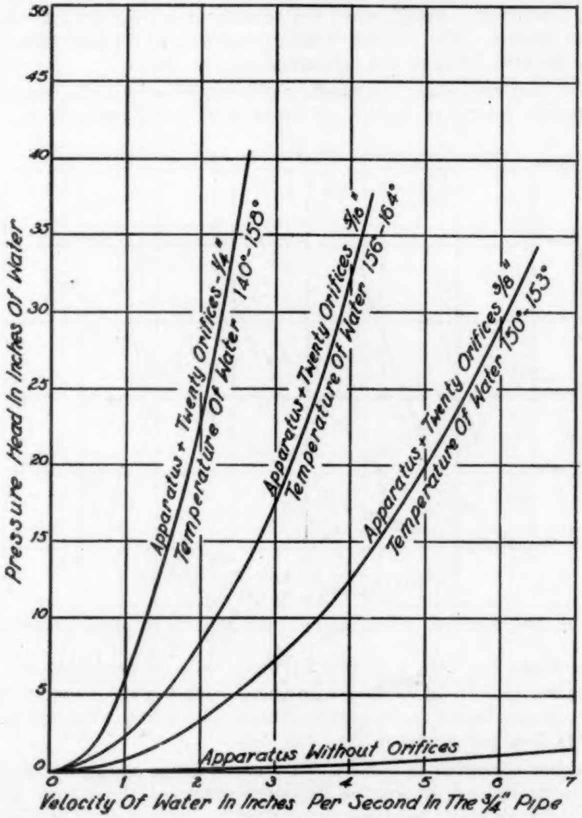


FIG. 13. HOT WATER FRICTION HEADS OF THE APPARATUS DEPICTED IN FIG. 10.

portion of this loss of head is produced by the 20 orifices and the remainder by the pipe line, AB.

To arrive at an approximate value of the loss of head caused by one orifice, let us assume that the water is flowing along the ¾ in. pipe with a velocity of 2 in. per second, and that it is forced to flow through a circular orifice and that, in flowing through the orifice, the stream lines converge, as indicated in Fig. 11, and as is known to be the case when the discharge through the orifice is into

the air, in which case, the area of the most contracted portion of the stream is about 62 per cent of that of the area of the orifice. When the stream of water is discharged into a body of water instead of into air it may be checked before its area can contract materially. Assuming that no contraction takes place, the maximum velocities at the three respective orifices will be 21.7, 13.87, and 9.66 in. per second, when the velocity of the water in the ¾ in. pipe is 2 in. per second. The respective heads required to produce these increases in velocity are 604, 240, and 115 mil-inches.

Assuming that the energy required to produce these velocity heads is transformed entirely into heat (which, in these cases, would elevate the tempera-

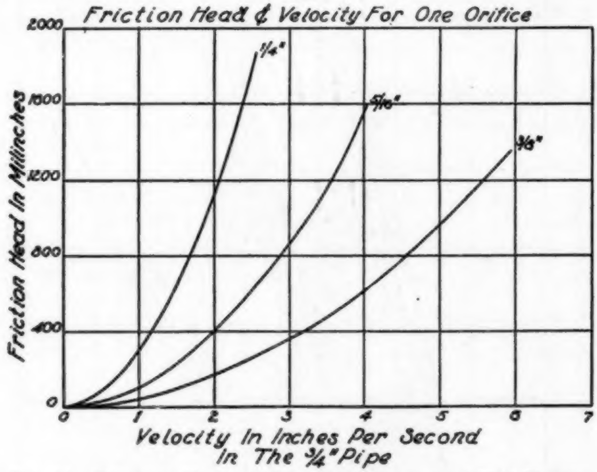


FIG. 14. FRICTION HEADS OF ORIFICES PLACED IN ¾-IN. UNIONS

ture of the water less than 1/10,000 F) these heads of 604, 240, and 115 mil-inches, respectively, would be the friction heads added to the radiator circuits by introducing a ¼, 5/16, or ⅜ in. orifice into a ¾ in. line when the water has a velocity of 2 in. per second in the ¾ in. line.

If the above reasoning is correct, the friction heads caused by such orifices will be practically independent of the temperature of the water, because, in these cases, the losses of head are not caused by internal fluid friction, but by transformation of kinetic energy into heat.

The results of the experimental determinations are shown in Figs. 12 and 13. The friction head caused by the pipe line *AB* (without the 20 orifice plates) is represented by the line near the bottom of the diagram. The friction heads caused by the pipe line *AB* plus the 20 orifices are shown by the three other lines which represent respectively the ⅜ in., 5/16, and ¼ in. orifices.

A comparison of these four lines shows that the friction head of the line *AB* without the orifices is so small when compared with the friction heads of the pipe line with the orifices that it may practically be neglected.

A comparison of the values shown by Fig. 12, which were secured with water having a temperature of 88 to 90 F, with the values shown by Fig. 13, which were secured with water having a temperature of 140 to 164 F, shows that the two sets of values are practically identical and that the friction head caused by

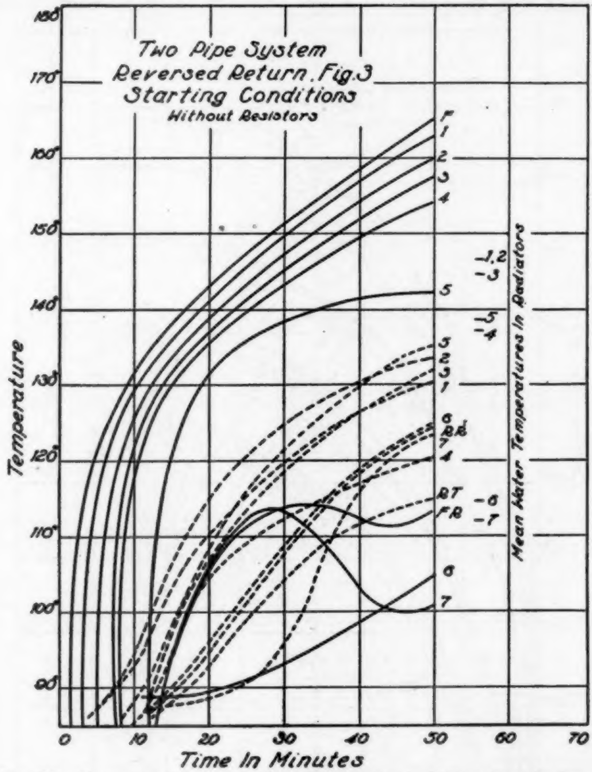


Fig. 15. Record of Starting Conditions for the Reversed-return System Without Resistors

an orifice is practically independent of the temperature of the water, as suggested above.

With the data of Fig. 13 as a basis, the curves of Fig. 14 were prepared for use in the design of hot-water heating systems.

Referring to Fig. 14, it will be seen that the friction heads of a 1/4 in., 5/16 in., and 3/8 in. orifice, for a velocity of 2 in. per second in the 3/4 in. pipe are, respectively, 1,130, 400, and 170 mil-inches or about 88 per cent, 66 per cent, and 46 per cent, respectively, more than those calculated on the assumption that there is no contraction of the stream flowing through the orifice. If these in-

creased friction heads are to be accounted for by a contraction of the stream through the orifice, the respective contractions in area, must be about 73 per cent, 77 per cent, and 82 per cent.

Having determined the friction heads caused by these orifice resistors, it is comparatively easy to determine which resistors and how many must be in-

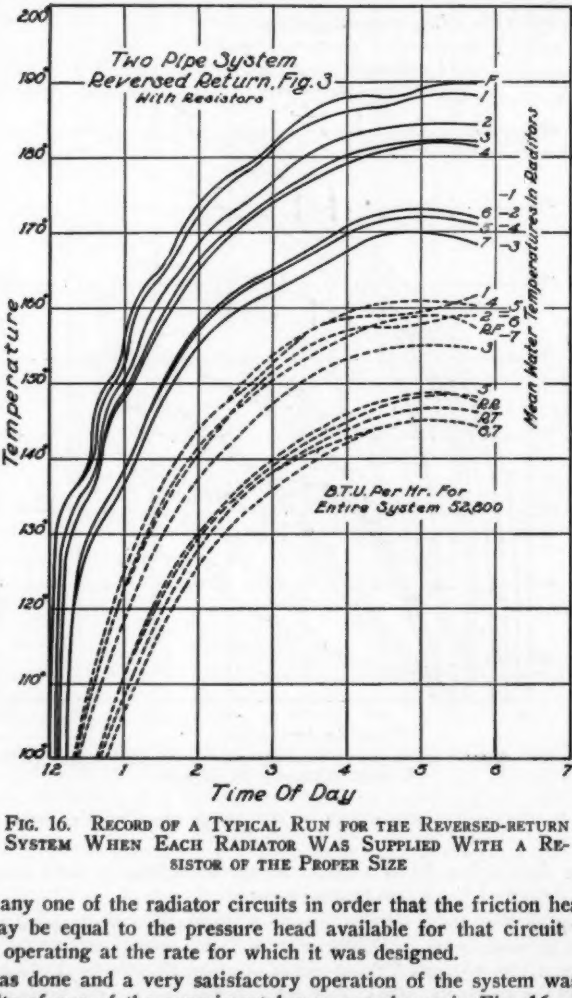


FIG. 16. RECORD OF A TYPICAL RUN FOR THE REVERSED-RETURN SYSTEM WHEN EACH RADIATOR WAS SUPPLIED WITH A RESISTOR OF THE PROPER SIZE

serted in any one of the radiator circuits in order that the friction head in that circuit may be equal to the pressure head available for that circuit when the system is operating at the rate for which it was designed.

This was done and a very satisfactory operation of the system was secured. The results of one of the experimental runs are shown in Fig. 16. It is evident, from this graph, that all radiators were operating in the right direction,

that the temperature drop through each radiator was about 25 F, and that the water entering Radiator 7 was almost 20 deg cooler than that entering Radiator 1.

This excessive cooling of the water before it reaches the last radiator empha-

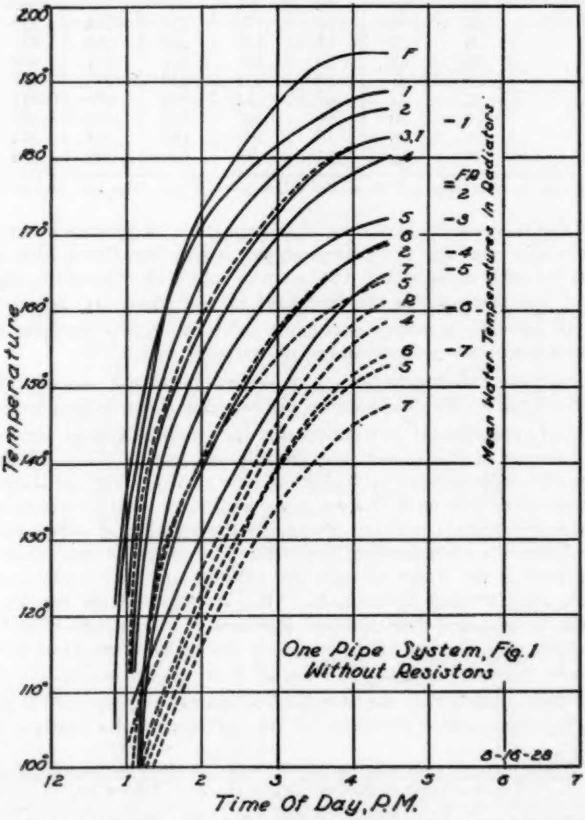


Fig. 17. Record of a Typical Run for the One-Pipe System Without Resistors

sizes the importance of covering the mains, or at least of covering the flow main.

Comparing the operation of the system as shown by Fig. 15, when the system had been very crudely designed according to approximate rules for pipe sizes, with the operation of the same system as shown by Fig. 16, when the design of the system had been improved by adding enough resistance to the several radiator circuits so that in every case the friction head for the circuit was equal to the pressure head for that circuit, will suggest that for a heating

TABLE 1. CAPACITIES IN 1,000 BTU OF MAINS FOR ONE-PIPE SYSTEMS FOR A TEMPERATURE DROP OF 35 DEG AND A HEAD OF 3 FT 6 IN.*

Length of Horizontal Mains	Pipe Sizes—Inches							
	1½	2	2½	3	3½	4	5	6
50	37	68	98	171	245	327	580	888
75	32	59	87	151	222	298	527	818
100	28	53	79	138	204	278	487	774
125	26	48	73	129	191	261	460	736
150	23	45	68	121	179	245	437	700
175	22	42	65	114	169	231	415	670
200	21	40	61	107	158	216	394	638
225	20	38	58	101	149	203	374	608
250	19	36	55	95	140	190	354	580

* If the head is more than 3 ft 6 in., the values given in this table may be increased.

system of the type under discussion it is impossible to prepare a set of tables from which pipe sizes can be selected in such a way that the system will function correctly, without including also a set of tables of resistors to supplement the table of pipe sizes, unless the pipe sizes for the mains are unduly large.

It may be possible to secure a radiator valve which will perform the functions of both the orifice resistor and of the radiator valve.

The calculations relating to the use of orifice resistors, mentioned above, are explained in detail in the discussion of the two-pipe direct-return system.

The second experimental installation was the one-pipe system, shown in Fig. 1. The operation of this system was entirely satisfactory. Fig. 17 shows one of the sets of results secured with the one-pipe system. Fig. 18 shows the results secured with the same system after additional friction heads had been introduced in the several radiator circuits by the addition of orifice resistors to the several risers as tabulated in Fig. 18. The effect of the resistors is to retard the flow of the water through the radiator and thereby to increase the temperature drop through the radiator. For example, in the run recorded in Fig. 17, the temperature drop through Radiator 2 is from 186 F to 169 F, or 17 F, whereas, with the use of the resistors, the temperature drop through the same radiator is from 195 F to 158 F, or 27 F, as shown in Fig. 18.

Fig. 17 shows clearly that the average temperature of the water in the radiators of a one-pipe system decreases as the distance of the radiator from the

TABLE 2. CAPACITIES IN 1,000 BTU OF MAINS FOR ONE-PIPE SYSTEMS FOR A DROP IN TEMPERATURE OF 25 DEG AND A HEAD OF 3 FT 6 IN.*

Length of Horizontal Mains	Pipe Sizes—Inches						
	2	2½	3	3½	4	5	6
50	41	59	100	140	195	346	552
75	35	51	90	130	179	320	500
100	31	48	83	121	165	293	450
125	30	44	75	113	153	280	425
150	27	40	70	104	143	265	405
175	25	38	66	98	136	252	390
200	24	35	63	93	130	239	377
225	23	34	61	90	125	225	367
250	22	33	60	88	120	223	358

* If the head is more than 3 ft 6 in., the values given in this table may be increased.

heater increases. Consequently, to secure a correct installation it is necessary to base the size of the radiator not only upon the number of Btu which the radiator is to dissipate, but also on the average temperature of the water in the radiator.

For the installation described herein, the main was not covered because it

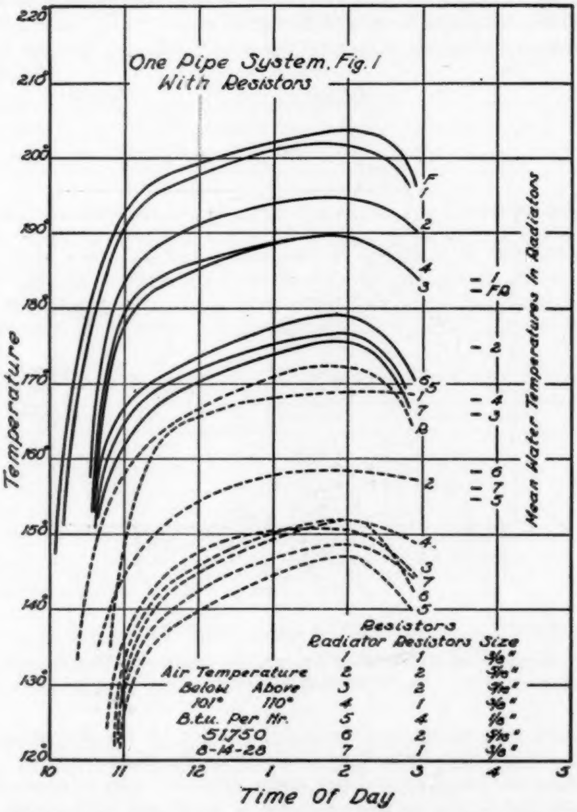


FIG. 18. RECORD OF A TYPICAL RUN FOR THE ONE-PIPE SYSTEM WHEN RADIATORS WERE EQUIPPED WITH RESISTORS

was felt that in most residences it is desirable to have the basement heated by the main and without the installation of basement radiators. When the basement need not be heated, the main should be covered to reduce the cooling of the water before it reaches the radiators.

A one-pipe system should be designed so that the drop in temperature through the system will be as small as practicable. This can be accomplished, in part, by using a large main and by covering the main.

The operation of the one-pipe system is so simple and so positive that it is comparatively easy to design and to install such a system, and it seems that it should be possible to prepare tables according to which pipe sizes and radiators can be determined with sufficient accuracy.

The following sets of tables and rules are proposed for this purpose.

Table 1 is based on a temperature drop of 35 deg through the system. Table 2 is based on a temperature drop of 25 deg.

Both tables are based on a head of 3 ft 6 in.; *i. e.,* on a vertical distance of

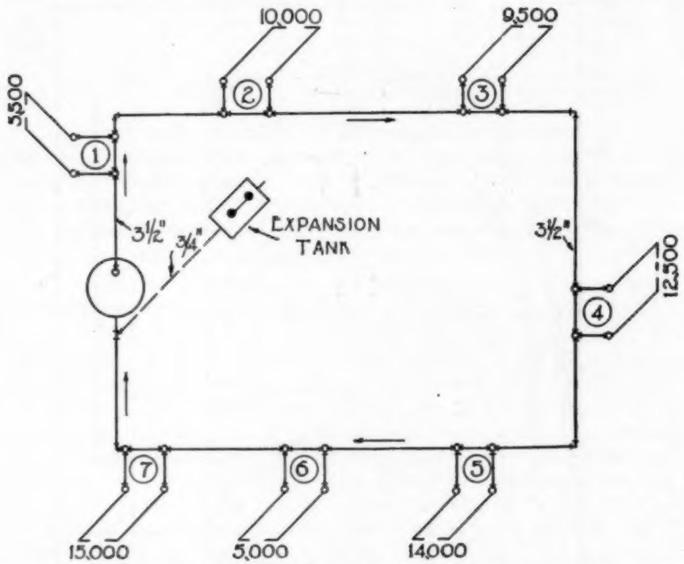


FIG. 19. DESIGN FOR A ONE-PIPE SYSTEM, ACCORDING TO TABLES AND RULES PROPOSED HEREWITH

3 ft 6 in. from the main to the center of the heater. If the distance is greater, the system will function better than indicated by the tables.

Both tables are based on the assumption that the main is covered. If the main is bare, the farther radiators will not be as efficient as the tables imply.

For both tables, the friction head calculations were based on the "Length of the Horizontal Main," listed in the tables, plus 10 ft of pipe and 10 elbow equivalents for risers and connections. These additions were assumed to be representative of the ordinary installations.

A comparison of Table 1 with Table 2 shows that the size of the main must be increased as the temperature drop through the system is decreased. For example, for a system delivering 200,000 Btu, with a length of horizontal pipe of 150 ft, a 4 in. main is required if the temperature drop is to be 35 deg, and a 5 in. main, if the temperature drop is to be 25 deg.

It will be shown below that, as the size of the main increases, the average size of the radiator decreases, thereby offsetting the increased cost of the main and conserving floor space.

Table 3 shows the capacities of risers for first, second, and third floor radiators. The table is based on the assumptions that the vertical distance of the first floor above the main is about 20 in., that the story heights are about 9 ft, that the individual resistances in every radiator circuit are approximately equivalent to 14 elbows, and that the temperature drop through each radiator will be 20 F or less.

Tables showing the capacities of risers supplying more than one radiator can be prepared but it is considered best not to propose such tables until after corresponding installations shall have been made and tested.

Rules for Determining Sizes of Radiators for One-Pipe Systems

The correct method of finding the sizes of radiators for one-pipe systems is to assume the total temperature drop through the system and, having done that, to calculate the successive partial drops in temperature which occur in the main at the several return risers. Knowing these partial temperature drops, find,

Table 3. Maximum Capacities in Btu of Risers for One-Pipe Systems*

Size, Inches	First Floor	Second Floor	Third Floor
¾	2,500	6,000	7,000
1	5,000	11,000	12,000
1¼	9,000	20,000	24,000
1½	12,000	....	....
2	21,000	....	....

* If valves are installed in the radiator circuits, they should be of the same size as the pipes. If smaller valves are used, they will introduce larger friction heads than were included in the calculations on which the table is based.

first, the temperature of the water in every flow riser, *second,* the average temperature of the water in the radiator, *third,* the value of k for the particular temperature difference—water to air,—and, *fourth,* the corresponding radiator size.

Using this method in designing a system in which there are 16 sets of risers, it will be necessary to make 16 sets of calculations. This method may be unnecessarily refined and it may be sufficiently accurate to use the following approximate method: Divide the system into four sections so that the radiators in each of the four sections deliver approximately the same quantity of heat to the building. Number the four sections consecutively in the order in which the water flows, so that Section 1 receives the hottest and Section 4 the coldest water. Calculate the sizes of the several radiators in the usual way and then multiply the calculated sizes by the factors shown in the following tabulation:

For a total temperature drop of	Section 1	Section 2	Section 3	Section 4
35 deg	1.00	1.15	1.30	1.45
25 deg	1.00	1.10	1.20	1.30

To illustrate the use of the tables and rules proposed above, let it be required to design the one-pipe system shown in Fig. 19.

There are 7 radiators delivering, together, 71,000 Btu per hour.

The length of horizontal main is 160 ft.

Let us adopt 25 F as the total temperature drop.

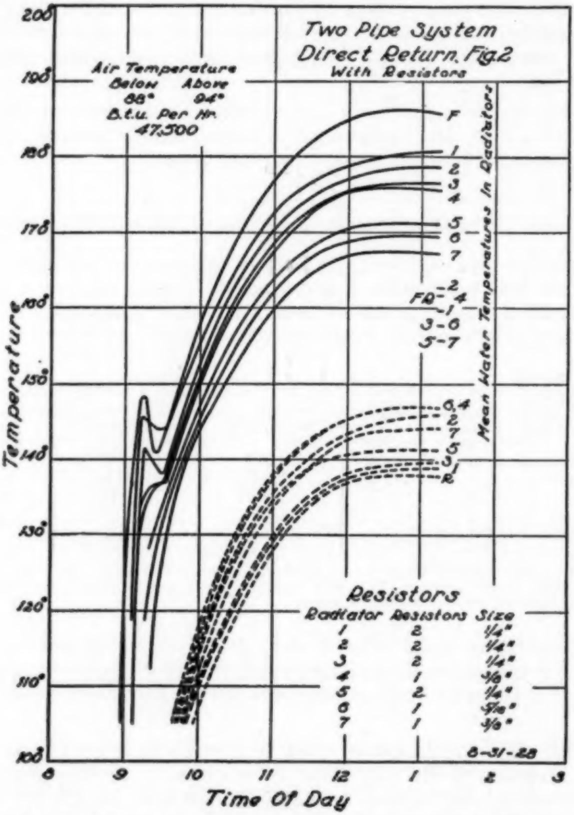


FIG. 20. RECORD OF A TYPICAL RUN FOR THE TWO-PIPE DIRECT-RETURN SYSTEM WHEN THE RADIATORS WERE EQUIPPED WITH PROPER RESISTORS

To find the size of the main: from Table 2, a 3 in. pipe is slightly too small and a 3½ in. pipe too large. We could construct the main partly of 3 in. and partly of 3½ in. pipe; or, if the main can be installed more than 3 ft 6 in. above the center of the heater, we could use a 3 in. pipe. It may be best to adopt the 3½ in. pipe in this case.

To find the sizes of the risers: from Table 3, the largest radiator, No. 7, hav-

ing a heat output of 15,000 Btu will need a 2 in. riser if located on the first floor, and 1¼ in. risers if located on the second or third floors. The smallest radiator, No. 6, having a heat output of 5,000 Btu, will need 1 in. risers if located on the first floor; and a ¾ in. riser if located on the second or third floor.

To find the sizes of the radiators: Let us assume that the water leaves the

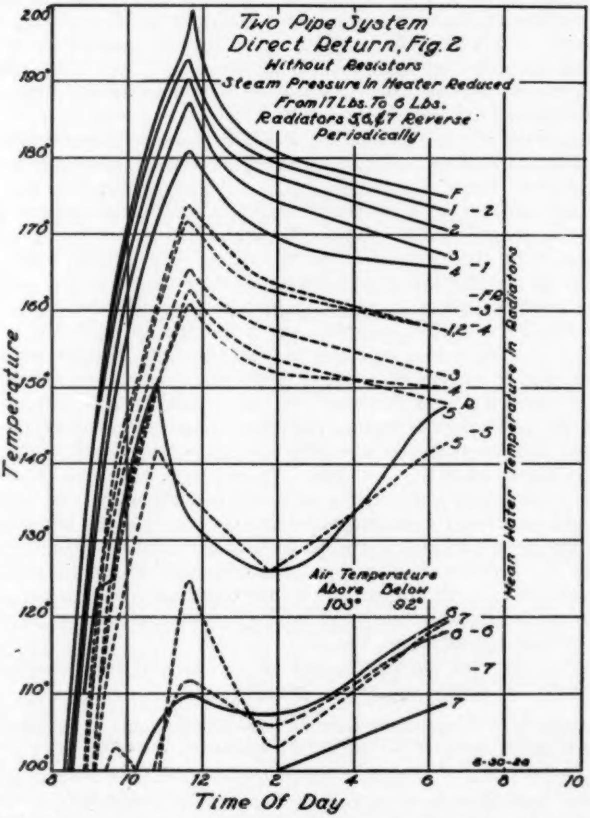


FIG. 21. RECORD OF A TYPICAL RUN FOR THE TWO-PIPE DIRECT-RETURN SYSTEM AFTER THE RESISTORS HAD BEEN REMOVED

heater at a temperature of 210 F when the system is delivering 71,000 Btu and that the radiators are operating with a temperature drop of 20 deg. The average water temperature in Radiator 1 will then be 200 F, and the temperature difference—water to air—will be 130 F. For this temperature difference, and for a 38 in. 3 column radiator, the value of *k* is 1.36, and the heat transmission is 1.36 x 130, or 176.8 Btu per square foot. Consequently, if all radiators re-

ceive water at the same temperature as Radiator 1, the radiator sizes would be as follows: No. 1, 28; No. 2, 57; No. 3, 54; No. 4, 71; No. 5, 79; No. 6, 28; and No. 7, 85 sq ft.

To divide the heating system into four sections so that each section delivers, approximately, the same quantity of heat; we would probably place Radiators 1 and 2 in Section 1, Radiators 3 and 4 in Section 2, Radiators 5 and 6 in Section 3, and Radiator 7 in Section 4. Multiplying the radiator sizes calculated above by the factors shown under the rules for determining radiator sizes, namely: 1.00, 1.10, 1.20, and 1.30, when the total temperature drop is 25 deg, we find the following sizes: No. 1, 28; No. 2, 57; No. 3, 59; No. 4, 78; No. 5, 95; No. 6, 34; and No. 7, 110 sq ft. These calculated sizes must be changed to the nearest stock sizes available.

The third experimental installation was the two-pipe direct-return system shown in Fig. 2. Having had the experiences described above with the earlier installations, this system was designed so that the friction head in the risers of each radiator was equal to the pressure head produced by the radiator plus that portion of the pressure head, produced by the heater, which was available for the radiator at the riser connections to the mains. The friction head was provided partly by the pipe line and partly by orifice resistors in the radiator circuit. The resulting operation of the system was very satisfactory as may be seen from Fig. 20, which is a record of one of the experimental runs.

As a final test, the orifice resistors described in Fig. 20 were removed and the system allowed to operate without them. The results secured are shown in Fig. 21, which seems to illustrate very nicely what really happens when a system is said to be *short-circuiting* and which proves conclusively that a system will not function correctly when the friction heads in some of the radiator circuits are too small.

In order to show how accurately a hot-water heating system can be designed, a careful check-calculation was made of the operation of the two-pipe direct-return system shown in Fig. 2, for the operating conditions shown in Fig. 20. The object of the check-calculation is to find whether or not the friction head for every radiator circuit is equal to the pressure head available for that radiator when the calculations are based on the actual conditions existing during the experimental run shown in Fig. 20.

The calculations are too complicated to be recorded here in detail. The general results for Radiator 1 are as follows:

Heater pressure head (Produced in main flow and return riser).....	649 mil-inches
Friction head in heater and in mains between heater and connections of risers of Radiator 1..	95 mil-inches
Remainder of heater pressure head, available to produce flow through Radiator 1	554 mil-inches
Radiator pressure head (Produced in radiator flow and return riser)	410 mil-inches
Total pressure head available for Radiator 1..............	964 mil-inches
Friction head in radiator and in connected pipe lines...............	50 mil-inches
Friction head in two ¼ in. orifice resistors........................	1,080 mil-inches
Total friction head in radiator circuit.....................	1,130 mil-inches

Excess of calculated friction head over calculated pressure head, 1,130—964, or 166 mil-inches.

Per cent error: 166/964 or +17 per cent.

Similar calculations were made for the other six radiators and the following general results were found:

Radiator	Calculated Pressure Head	Calculated Friction Head	Per cent Error
1	964	1,130	+17
2	835	920	+10
3	780	844	+ 8
4	590	515	—13
5	1,729	1,037	—40
6	1,322	1,214	— 8
7	379	375	— 1

It is evident from these check calculations that every radiator in the system is functioning as would be predicted from a correct design, except Radiator 5.

The calculated friction head in the circuit of this radiator is about 692/1729, or 40 per cent smaller than the actual friction head. No explanation for this

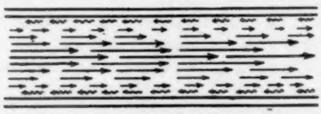


FIG. 22. POSSIBLE DOUBLE FLOW, CAUSED BY COOLING DURING LOW VELOCITIES, IN A FLOW RISER, PRODUCING AN EXCESSIVE FRICTION HEAD

large discrepancy could be found unless it be that the velocity of the water in the flow riser is so low that the water in contact with the pipe cools sufficiently to reverse the direction of its flow along the outer portion of the stream, as sketched in Fig. 22. If such a condition should exist it might account for the high friction head which was found in the actual installation and it would suggest the existence, in the flow risers of hot-water heating systems, of a "critical velocity" below which the system will not function correctly.

The subject of a *critical velocity* in hot water heating pipes may be very important and well worth careful study as a separate research project, in connection with additional studies, similar to those described in this paper, which will be necessary before tables can be safely proposed that may be used in the design of two-pipe heating systems.

In conclusion, the authors express their appreciation of the valuable assistance given them in this work by G. H. Glover and W. H. Badgett, seniors in Texas Agricultural and Mechanical College, especially in the preparation of diagrams and with the check-calculations.

DISCUSSION

H. H. ANGUS: In Canada we use hot water heating to a large extent for residences, hospitals, boarding schools and similar buildings. On the larger buildings we find it advisable to use a pump to increase circulation. These pumps usually operate at a very low head, in many cases 6 in. or less, and the power required is very small. Due to friction in glands and bearings, however, it is not advisable to use a motor of less than ¼ hp.

The advantage of the pump is, that you get circulation more quickly than by gravity, so that, on a job designed for 30 deg drop with gravity you will get probably 5 to 10 deg drop when using the pump. This means that in cold weather the temperature of the radiators is considerably higher than it would be with gravity circulation also when the fire is increased the radiators respond very quickly. The pump also will generally improve very considerably the circulation on a job which is poor and will provide uniform circulation on a system where some radiators are hard to heat.

On oil burning furnaces the thermostat controlling the oil burner is generally placed near the centre of the building and it shuts off the burner sometimes before radiators at the far ends have had time to heat up. By wiring the pump so that it operates at the same time as the oil burner the heat absorbed by the water is carried to all sections of the building uniformly.

On the medium sized jobs where pumps of this kind are used, it is customary to use only one pump and as this may get out of order at certain times, the system should be designed with sufficiently large pipes to allow for enough circulation to prevent discomfort. Therefore while the pipe sizes may be considerably smaller than for a 30 deg gravity job it is advisable to keep them large enough for a 50 or 60 deg gravity job to provide for emergency.

I understand Professor Giesecke is doing some work on pipe sizes for systems of this kind as there is considerable room for improvement in the ordinary gravity system.

HOMER R. LINN: I think Mr. Angus mentioned an essential point in his comments on oil burner installations for hot water heating. I have found that when a pump is installed in the line a lot of troubles that have been charged against the oil burner are corrected; particularly that of not getting the water to the radiator before the burner shuts off. One question I wanted to ask Professor Giesecke is whether in his one-pipe system he determined the proper distance between the flow and return connections to the main in proportion to the size of the radiator?

H. M. NOBIS: I notice that Professor Giesecke used a steam generator. I believe that the use of a coal fired boiler would produce different circulating speeds, due to the fact that by higher fuel temperatures a steam bubble liberation takes place which facilitates the movement of the water within the system. I also notice that the radiators are all connected on the bottom, especially the flow line and I would like to point out that if you supply hot water at the bottom of a radiator it gets cooler immediately because of the mixing that takes place with the colder radiator water.

R. C. BOLSINGER: I would like to ask if in making the connections in the one-pipe system, there is any difference between using the partition fitting or just a regular fitting.

PROFESSOR F. E. GIESECKE: Regarding the question of a pump to increase pressure heads: This is a question of relative economy, namely, is it cheaper to install, maintain, and operate a pump, or to install pipes of sufficient size to secure satisfactory gravity circulation? A calculation should be made to secure the correct answer.

Regarding Mr. Linn's question: We have made no tests to determine the safe minimum distances between radiator connections to the main. I believe, for gravity circulation systems, it is sufficient to make the distances equal to those between the respective risers. For forced circulation systems, it may be necessary to use greater distances because the water has a higher velocity in the main and, consequently, a larger pressure head is necessary to divert a sufficient quantity of water from the main into the flow risers. For such systems, the proper distances can be calculated; they depend, in part, upon the relative sizes of the main and the risers.

Regarding the difference between the riser connections to the upper and the lower radiator tappings: Tests have been conducted at the University of Illinois to determine this difference. The radiator used was a 3-column, 38-in. radiator. When the upper tapping was used, the hot water distributed itself uniformly along the upper portion of the radiator and then flowed uniformly downward through all sections of the radiator. When the lower tapping was used, the hot water moved upward through the middle one of the three columns, and then downward through the outer columns. The heat dissipated by the two radiators was practically the same but there was a considerable difference between the paths followed by the water in flowing through the two radiators. It is possible that, with other types of radiators, the riser connection at the lower tapping may be less efficient than it was found to be with the 3-column radiator tested at the University of Illinois.

In our own tests, as explained before, we first used the lower connections and later changed to the upper connections; the change made the operation of the system worse than it had been; with the lower connections we had only one radiator running backward; with the upper connections we had two radiators running backward. The change increased the radiator pressure head unduly as compared with the heater pressure head.

I do not know how important or how desirable it is to use partition fittings.

President Lewis: Professor Giesecke certainly keeps us on our mental toes to follow him, but he has reduced his idea to fundamentals and he has made a complicated matter seem simple. I think he deserves a lot of credit for that conception of pressure differential which he has just explained.

S. R. Lewis: Mr. President, I cannot forbear to mention my pleasure about the success of the cooperative agreement with the Texas Agricultural and Mechanical College. Professor Giesecke is bringing hot water back to us.

We must not forget what a wonderful tool hot water is in heating. I recall a case of recent experience. An architect said, "I have a house on a hillside in Glencoe and a planned addition brings the boiler room up about the second floor level. I have a billiard room and other occupied rooms down on the hillside in the basement. I do not want any pipes under the floor nor around the walls, and the problem is to properly heat that house."

An oil burning hot water boiler with an electric booster pump was installed and the pipes were run overhead and down the walls, with the radiators on the floors. A very nice letter from the architect said: "You have solved it; the hot-water plant with the pump does the business."

I am for hot water heating and I am for Professor Giesecke, who produces the goods. He was one of the first men to respond to the Guide Publication Committee and send in a revised chapter on hot water pipe sizes and hot water heating for THE GUIDE 1930.

No. 840

OVER-ALL HEAT TRANSMISSION COEFFICIENTS OBTAINED BY TESTS AND BY CALCULATION

The results of cooperative research between the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the University of Minnesota

By FRANK B. ROWLEY,[1] A. B. ALGREN,[2] J. L. BLACKSHAW,[3] MINNEAPOLIS, MINN.

ONE of the important factors in the problem of heat transmission through building materials and insulated walls is that of the relation between the over-all coefficients as obtained by test and as determined by calculation for a built-up wall section. There are two coefficients commonly used to designate the thermal value of materials: the over-all coefficient, and the thermal conductivity of the material. The first is generally designated by U and gives heat transmission in British thermal units per hour per square foot of wall per degree difference in temperature between the air on the two sides of the wall. The second is designated by k and gives the thermal conductivity of the material in British thermal units per square foot per hour per inch thickness per degree difference in temperature between the two surfaces. The difference between these coefficients is that the first one covers the full thickness of the wall giving the transmission from air to air, while the second one gives the transmission from surface to surface per inch thickness of the material. The first is applicable to built-up wall sections, and the second to the conductivity of homogeneous materials.

It is conceded that the conductivity, k, for a material is much more easily determined by test methods than is the over-all coefficient.

In the first place, the apparatus commonly used is much less expensive to construct and the results may be obtained at a much less expense of time and material. The over-all coefficient is, however, the one necessary for finished buildings and it must either be determined by test or indirectly by calculation. If it is determined by calculation, the results must be sufficiently checked by tests to prove the correctness of the procedure.

The method of procedure and the type of apparatus for determining the thermal conductivity of homogeneous materials is fairly well standardized and accepted by the various experimenters. This consists, in general, of a hot and a cold plate between which the test material is placed, the hot plate being so arranged that the heat to the test specimen may be measured and the heat

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losses prevented by a spcial guard or balancing ring. The method for determining the over-all coefficients is not yet as commonly accepted. In general, there are two procedures possible; first by an actual test of the built-up wall section, and second, by calculation. Several types of test apparatus have been proposed and used. One type consists of some especially designed box and the other is some form of plate or meter placed in contact with the surface of the wall. In any case, if the results are to be of value, great care is necessary to prevent any unknown loss of heat and to obtain accurate measurements of temperatures and heat flow.

If the over-all coefficient is to be determined by calculation, there are several factors pertaining to the specific materials used in the walls which must be definitely known. Without these the results may be incorrect and misleading.

Some of the factors which enter into the calculation are as follows:

A. The thermal conductivities of the individual materials used.
B. The arrangement of these materials in the structure.
C. The conductivity of the air spaces within the wall.
D. The surface coefficients of the outer layer of the materials.

"A" The thermal conductivities of the materials are normally determined by the hot plate method. They depend upon the character of the material, density, moisture content and mean temperature. The conditions governing these values must all be considered if the conductivities are to be applied with accuracy to any particular wall. A common mistake is to use conductivities which were obtained for an entirely different mean temperature, moisture content or density than those at which the material is to be used in the wall.

"B" In considering the arrangement of materials, in the wall, such factors enter as the proximity of these materials to each other, the thickness of the particular material in the construction as compared to the thickness when tested, the blocking off of the air spaces and the occurrence of air circulation through the materials themselves. As examples of possible errors, a flexible, soft material which is rated at ½ in. thickness may be nailed between two layers of boards and brought down to less than ¼ in. in thickness. Materials which are porous in nature may be placed in a wall in such a manner that natural air circulation through the material will offset a large part of the insulating value. The application of the material may be such that heat is transferred around it by air currents. When any of these factors are present, the results obtained by calculation are misleading.

"C" The values assigned to the air spaces will vary depending primarily upon the thickness of the air space and the mean temperature. A common method has been to assign one value for all air spaces, which, in the light of recent experimental work, is incorrect. In selecting these values, it must be definitely known that the air space is a closed compartment and not open to communication with other air spaces of different temperatures.

"D" When considering the inner and outer surface coefficients of the wall, due regard must be given to the character of the surfaces, the nature and temperature of the surrounding objects and the wind velocity to be expected over the surfaces. For practical results the wind velocity is the most important factor to be considered.

From the foregoing it is evident that there are many elements which enter

into the method of determining over-all coefficients by calculation. If, however, these elements are all definitely understood, and there are no other unknown factors as infiltration entering into the problem, the over-all coefficients may be calculated with accuracy, and the results will check with those obtained by accurate test methods. There are cases, however, for which calculations are difficult, due to complications of the structure, to air leakage, to changes in the form of material when applied, to poor workmanship, etc. In such cases these conditions must either be estimated for the calculations or else the constants must be determined by test.

There has been much controversy over the relative merits of the two general methods for determining the over-all coefficients. Some have advocated the test method and others that the constant be determined by calculation.

It is easy to find plenty of examples where both methods have given erroneous results. For this reason some engineers have taken the stand that there is no acceptable method for determining these over-all coefficients. In spite of the controversies which may exist, the fact is that accurate results may be obtained by either method when properly applied. If test methods are used, accurate instruments and methods must be employed, and if calculation is resorted to, the full conditions covering the specific properties of the materials and their application to the wall must be known.

It is not the purpose of this paper to discuss the relative merits of different types of test apparatus which have been used for determining over-all heat transmission coefficients, but rather to consider the results as obtained by one specific type and to compare these to the results obtained for the same walls by analytical methods. The hot box apparatus was used to determine the over-all coefficients and the hot plate to determine the conductivities of the individual materials. These were described in a paper entitled Heat Transmission Research and published in the TRANSACTIONS OF THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, 1928, p. 439. The description of the apparatus will, therefore, be omitted.

For calculations the following well known formula was used:

$$U = \frac{1}{\frac{1}{f_1} + \frac{x_1}{k_1} + \frac{1}{a_1} + \frac{1}{a_2} + \frac{x_2}{k_2} + \frac{1}{f_0} + \cdots}$$

in which U is the over-all heat transmission coefficient of the wall, f_1 and f_0 are the inner and outer surface coefficients of the wall, x_1 and x_2, etc., are the thicknesses of the various materials used in the wall, k_1 and k_2, etc., are the respective conductivities of the materials, a_1 and a_2 are the air space coefficients for the air spaces as occurring in the walls.

These factors were determined for the walls considered as follows. The conductivity of all insulating materials used was determined by the hot plate method. The conductance was determined for all combinations of materials as were used on the surfaces of the walls. Thus lath and plaster; sheathing, building paper and lap siding; sheathing, building paper and stucco were treated as separate units and the conductance determined for the mean temperatures at

TABLE 1—DESCRIPTION OF INSULATION USED IN THE DIFFERENT WALLS

Mark Designating Material	Description of Material	Wall in Which Used	Thickness as Used Inches	Density Lb per Cu Ft	Conductivity k	Mean Temp F
A	Semirigid board	19	0.580	13.3	0.297	35.0
		36	0.660	....	0.300	40.0
B	Wood fiber board	10 outside	0.520	15.2	0.315	30.0
		11 inside	0.526	14.4	0.330	62.0
		11 outside	0.527	15.1	0.315	30.0
		34	0.526	15.0	0.318	40.0
C	Felted wood fiber between two layers of craft paper	12	0.750	4.8	0.253	40.0
		16	0.625	4.8	0.250	35.0
		53D	0.750	4.8	0.253	40.0
E	Animal hair lined on one side with tar paper and on the other with heavy craft paper	15	0.36	9.7	0.238	40.0
G	A cereal fiber board	23 inside	0.48	14.4	0.362	62.0
		23 outside	0.47	14.8	0.340	22.0
		26 inside	0.48	14.4	0.362	62.0
		26 outside	0.47	14.8	0.342	30.0
H	Paper felt treated on the surfaces with creosote for water-proofing	28	0.119	34.4	0.470	40.0
J	A ground waste paper pulp held together with a binder and applied by air gun	33 outside	1.11	5.78	0.272	35.0
		37 outside	1.11	5.78	0.272	35.0
		42	0.814	5.78	0.283	60.0
K	Corrugated paper board	8B	0.209	11.7	0.374	40.0

TABLE 2—CONDUCTANCE OF SPECIFIC COMBINATIONS OF MATERIAL USED FOR CALCULATING THE OVER-ALL TRANSMISSION COEFFICIENTS OF WALLS

Materials	Mean Temperature	Conductance
3/8" lath and 3/8" plaster	70	2.50
Fir sheathing, building paper and pine lap siding	20	0.50
Fir sheathing, building paper and stucco	20	0.82
Fir sheathing and building paper	30	0.706
Building paper and 4" pine lap siding	15.5	0.854
Conductance of 3/8" plaster	73	8.8

which they occurred in the walls. The air space coefficients were taken from the curves as previously determined and reported in a paper entitled Thermal Resistance of Air Spaces, TRANSACTIONS OF THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, 1929, p. 165. The surface coefficients were taken as the average results obtained from many tests with the hot box. These surface coefficients are for an air velocity of about 0.6 mph and place the calculated results on the same basis as the test results.

The conductivities of the insulating materials used as determined by the hot plate for the mean temperatures corresponding to their position in the walls is given in Table 1. The conductance of the specific combinations of materials for mean temperatures at which these combinations occurred is given in Table 2. The surface coefficients as determined from tests and used in the calculations are given in Table 3. The air space coefficients as previously determined were taken from the curves of Fig. 1. A complete description of the walls tested and calculated is given in Table 4. The individual coefficients and factors used in calculating the over-all coefficient for each wall are given in Table 5. The majority of these values were taken from Tables 1, 2 and 3 and the curves of Fig. 1, and compiled in Table 5 as a matter of convenience. The final results giving the relation between the calculated and test values are given in Table 6. It should be noted that in cases where a soft felted insulation was used the thickness was determined as accurately as possible after the material was in place in the wall. In the case of wall No. 15 this actually introduced an additional air space over part of the surface of the insulating material.

In selecting the values for the conductance of air space in framewalls, the question arises as to what insulating value should be given to the studding

TABLE 3—SURFACE COEFFICIENTS USED FOR CALCULATING THE OVER-ALL TRANSMISSION COEFFICIENTS OF WALLS

Surface	Mean Temperature	f_i	f_o
Plaster	77	1.8	....
Sheet Rock	75	1.7	....
Stucco	5	....	1.5
Siding	5	....	1.6

TABLE 4—DESCRIPTION OF WALLS TESTED AND CALCULATED

Wall No.	Type of Wall	Inside Construction	Outside Construction	Insulation	Air Space
8	Frame	3/8" wood lath ...... 5/8" plaster	3/4" fir sheathing, building paper 4" pine lap siding	None	One 3½" air space.
8A	Frame ...... Same as 8 (Note 1)	3/8" wood lath ...... 5/8" plaster	3/4" fir sheathing, building paper 4" pine lap siding	None	One 3½" air space.
8B	Frame ...... (Notes 2, 3)	3/8" wood lath ...... 5/8" plaster	3/4" fir sheathing, building paper 4" pine lap siding	Two thicknesses of insulation K applied between studding to divide air space into three parts	Two 1⅓" air spaces. One ⅞" air space.
10	Frame	3/8" wood lath ...... 5/8" plaster	Insulating board B............ Building paper .... 4" pine lap siding	None	One 3½" air space.
11	Frame	Insulating board B.. 5/8" plaster	Insulating board B............ Building paper .... 4" pine lap siding	None	One 3½" air space.
12	Frame ...... (Note 2)	3/8" wood lath ...... 5/8" plaster	3/4" fir sheathing, building paper 4" pine lap siding	Insulation C flanged midway in air space between studding...	Two 1⅝" air spaces.
15	Frame ...... (Notes 2, 4)	3/8" wood lath ...... 5/8" plaster	3/4" fir sheathing, building paper 4" pine lap siding	Insulation E flanged midway in air space between studding...	Two 1½" air spaces.
16	Frame	3/8" wood lath ...... 5/8" plaster	3/4" fir sheathing, building paper 4" pine lap siding	Insulation C nailed on studding under sheathing	One 3½" air space.
19	Frame	3/8" wood lath ...... 5/8" plaster	3/4" fir sheathing, building paper 4" pine lap siding	Insulation A placed back against sheathing	One 3" air space.
23	Frame	Insulating board G nailed to studding.	Insulating board G nailed to studding	None	One 3½" air space.
26	Frame ...... (Note 5)	Insulation G .... 5/8" plaster	Insulation G, building paper.... 4" pine lap siding	None	One 3½" air space.

28	Frame ...... (Note 2)	⅜" wood lath ...... ⅝" plaster	¾" fir sheathing, building paper 4" pine lap siding	Two thicknesses insulation H spaced between studding to divide air space into three equal parts	Three 1.08" air spaces.
33	Frame	⅜" wood lath ...... ⅝" plaster	¾" fir sheathing, building paper 4" pine lap siding	1.11" of insulation J applied between studding directly to sheathing	One 2.4" air space.
34	Frame ...... (Notes 2, 6)	⅜" wood lath ...... ⅝" plaster	¾" fir sheathing, building paper 4" pine lap siding	Insulation B, with heavy craft paper on cold side, placed midway between studding	Two 1½" air spaces.
36	Frame ...... (Note 2)	⅜" gypsum .... Papered on each side.	¾" fir sheathing, building paper 4" pine lap siding	Insulation A, covered on both sides with light craft paper..	Two 1⅜" air spaces.
37	Frame with brick veneer. (Note 7)	⅜" wood lath ...... ⅝" plaster	Face brick	Insulation J, 1.11" thick, applied between studding directly to sheathing	One ¾" air space between brick and paper in addition to air space of wall 33.
42	8" brick .... (Note 8)	⅜" wood lath ...... ⅝" plaster	4" face brick backed with 4" common brick	Insulation J, .814" thick, applied to inner surface of brick between 1⅝" furring strips.....	One .5" air space between insulation and lath. One .6" air space between inner and outer courses of brick.
53D	Frame ...... (Note 2)	⅜" gypsum .... Papered on each side.	¾" fir sheathing, building paper 4" pine lap siding	Insulation C flanged between studding	Two 1⅜" air spaces.
53F	Frame	⅜" wood lath ...... ⅝" plaster	¾" fir sheathing, building paper 4" pine lap siding	None	One 3½" air space.

Notes for Table 4:
(Note 1) Wall 8A Test area of wall was blocked off to prevent convection currents.
(Note 2) 40 F was taken as the average mean temperature for the air spaces.
(Note 3) Wall 8B A paper board header was used in air space at three-foot intervals.
(Note 4) Wall 15 The application of the insulating material causes air spaces to be formed between the craft paper and hairfelt. These spaces cover about 65 per cent of the area and average 0.6 in. in thickness.
(Note 5) Wall 26 Wall 26 was made up by adding surface finishes to wall 23.
(Note 6) Wall 34 Air space between insulating board and craft paper averaged 0.2 in.
(Note 7) Wall 37 The siding was removed from Wall 33. A 4-in. face brick was then added leaving a ¾-in. air space between brick and sheathing. The conductivity for the brick was taken from a test on a 4-in. clay brick wall.
(Note 8) Wall 42 The inner surface of brick was covered with two coats of waterproof cement before the insulation was applied.

which passed through this air space. From a practical standpoint, this would naturally be neglected, but for the purpose of more accurate checks, this effect should at least be considered. The thermal resistance of the studding may be easily determined and the average resistance for air space and studding might be calculated on the basis of parallel circuits of heat flow. The results thus ob-

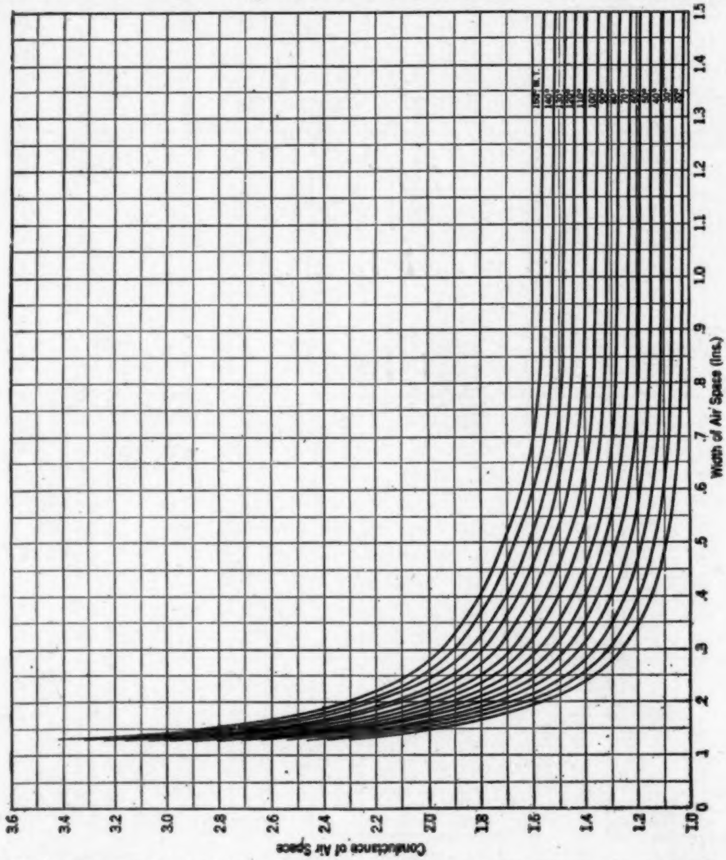

Fig. 1.—Conductance of Air for Different Mean Temperatures Fahrenheit

tained would only be true, in case the studding did not effect the conductivity of the air space. If we analyze the condition, it is evident that the greater part of the heat transferred across the air space is by the convected air currents, the air taking the heat from the hot surface transferring it over to the cold surface. The effect of the surface of the studs which pass through the air space is to increase the hot and cold surfaces in contact with the air, or in other words, some of the heat transferred across the space is picked up from the sur-

TABLE 5—COEFFICIENTS USED FOR CALCULATING THE OVER-ALL TRANSMISSION CONSTANTS OF WALLS

Wall No.	f_i	Mean Temp. F	f_o	Mean Temp F	Conductance Inner Construction	Mean Temp. F	Conductance Outer Construction	Mean Temp F	Conductivity Insulation	Mean Temp F	Conductance of Air Space	Mean Temp F
8	1.8	77	1.6	5.0	2.50	70	0.50	20.0	None		1.14	48.0
8A	1.8	77	1.6	5.0	2.50	70	0.50	20.0	None		1.14	48.0
8B	1.8	77	1.6	5.0	2.50	70	0.50	20.0	0.374	40.0	(3) 1.10	40.0
10	1.8	77	1.6	5.0	2.50	70	0.354	26.0	0.315	30.0	1.14	48.0
11	1.8	77	1.6	5.0	0.587	63	0.354	26.0	0.330 Inside	62.0	1.10	40.0
									0.315 Outside	30.0		
12	1.8	77	1.6	5.0	2.50	70	0.50	20.0	0.253	40.0	(2) 1.10	40.0
15	1.8	77	1.6	5.0	2.50	70	0.50	20.0	0.238	40.0	(2) 1.10	40.0
16	1.8	77	1.6	5.0	2.50	70	0.50	20.0	0.250	35.0	1.14	48.0
19	1.8	77	1.6	5.0	2.50	70	0.50	20.0	0.297	35.0	1.14	48.0
23	1.7	77	1.6	5.0	0.753	62	0.723	22.0	0.362 Inside	62.0	1.10	40.0
									0.340 Outside	22.0		
26	1.8	77	1.6	5.0	0.696	63	0.392	26.0	0.362 Inside	62.0	1.10	40.0
									0.342 Outside	30.0		
28	1.8	77	1.6	5.0	2.50	70	0.50	20.0	0.470	40.0	(3) 1.10	40.0
33	1.8	77	1.6	5.0	2.50	70	0.50	20.0	0.272	35.0	1.14	48.0
34	1.8	77	1.6	5.0	2.50	70	0.50	20.0	0.318	40.0	(2) 1.10	40.0
											(1) 1.62	37.0
36	1.7	75	1.6	5.0	3.30	70	0.50	20.0	0.300	40.0	(2) 1.10	40.0
37	1.8	77	1.6	5.0	2.50	70	(4" brick 1.24)	19.0	See wall 33		1.14	48.0
											1.04	20.0
42	1.8	77	1.6	5.0	2.50	70	(4" brick 1.24)	19.0	0.283	60.0	(1) 1.32	68.0
											(1) 1.10	30.0
53D	1.7	75	1.6	5.0	3.30	70	0.50	20.0	0.253	40.0	(2) 1.10	40.0
53F	1.8	77	1.6	5.0	2.50	70	0.50	20.0	None		1.14	48.0

face of the studding next to the hot side and given back to the surface of the studding next to the cold side of the space. The effect of the studding on increasing the heat transfer in this manner depends upon their conductivity, and upon the ratio of the area of the exposed surfaces of the studding to the width of the air space. It is evident that the exact relation here would be very difficult to determine, but that the effect would be to increase the heat transfer across the air space and to neutralize the effect due to the additional insulating value of the studding. When it is considered that only 10 per cent of the air space area is covered by the studding, it is evident that the effect is very small, and in cases where insulation is placed in the air space between the studding, the insulating value of the air space and studding is substantially the same and any possible difference may be neglected. Tests made

TABLE 6—RELATION BETWEEN CALCULATED AND TEST RESULTS

Wall No.	Calculated Value of U At 40 F Mean Temp	Hot Box Value of U At 40 F Mean Temp	Percent Variation Based on Test Results
8	0.224	0.226	0.88
8A	0.224	0.225	0.45
8B	0.134	0.139	3.59
10	0.189	0.186	1.61
11	0.151	0.153	1.31
12	0.119	0.115	3.48
15	0.134	0.134	0.0
16	0.144	0.141	2.13
19	0.155	0.155	0.0
23	0.207	0.212	2.36
26	0.165	0.168	1.79
28	0.146	0.142	2.82
33	0.117	0.116	0.86
34	0.131	0.129	1.55
36	0.133	0.130	2.31
37	0.103	0.106	2.83
42	0.136	0.137	0.73
53D	0.120	0.120	0.0
53F	0.224	0.217	3.22

for the purpose of determining air space values with and without studding passing through the test area showed that there was no appreciable difference in the over-all coefficients. The effect of the studding was, therefore, omitted in all calculations for the over-all coefficients.

An analysis of the results given in Table 6 shows that there is a very close agreement between test and calculated results, the maximum variation being 3.59 per cent. For wall 8B which shows the greatest variation, the insulation was held between the studding by friction and there was a possibility of a small amount of infiltration of air past the joints which would increase the test results. Wall 12 was insulated with a felted material between two layers of paper in which there was some uncertainty as to the exact thickness of the insulation. Some variation may also be accounted for by the different characteristics of walls built after the same specifications. For instance, several of the walls have been duplicated from different lots of material. The average variation in test results has been found to be 2 per cent with a maximum of 5 per cent. The sheathing of the wall which showed a 5 per cent variation was found to be

TABLE 7—TESTS SHOWING IMPORTANCE OF CONSIDERING POROSITY AND APPLICATION OF INSULATION WHEN CALCULATING HEAT TRANSMISSION CONSTANT OF A WALL

Wall No	Type	Inside Construction	Outside Construction	Special Insulation	Hot Box Coeff. of Heat Trans. U at 40 F Mean Temp.	Calcul. Coeff. of Heat Trans U at 40 F Mean Temp.
53A	Frame	⅜" gypsum Papered on each side.	¾" fir sheathing, building paper 4" pine lap siding...........	None	0.220	0.228
53B	Frame	⅜" gypsum Papered on each side.	¾" fir sheathing, building paper 4" pine lap siding...........	A porous material placed midway in air space between studding	0.154	0.133
53C	Frame	⅜" gypsum Papered on each side.	¾" fir sheathing, building paper 4" pine lap siding...........	Same as 53-B with 1" of material removed from top and bottom of wall section between studding	0.177	0.133
53D	Frame	⅜" gypsum Papered on each side.	¾" fir sheathing, building paper 4" pine lap siding...........	An impervious material placed midway in air space between studding	0.120	0.120
53E	Frame	⅜" gypsum Papered on each side.	¾" fir sheathing, building paper 4" pine lap siding...........	Same as 53D with 1" of material removed from top and bottom of wall section between studding	0.184	0.120

about 3/32 in. thicker than that on the walls used for comparison. Such variations must be expected in any type of construction.

The results likewise show that accurate results cannot be expected by analytical methods, unless due regard is given to the selection of the individual constants which enter into these calculations. The values of the individual materials are affected by several variables and their selection must be governed by the conditions under which the materials are used in the wall.

For walls in which materials are impervious to the flow of air and are of definite thickness and location, the calculations are comparatively simple. For conditions in which the materials are porous or where the thickness in the wall is indeterminate, with indefinite air spaces and possible communication between the air spaces, the calculated results are less accurate and more apt to be misleading. In order to determine the effect of some of these doubtful conditions, a wall was built with a removable inner surface. This wall was tested without insulation and with different materials applied between the studding. The results of these tests are given in Table 7. The original wall without insulation, or 53A, gave results by test and by calculation which check very closely. For wall 53B with a porous material placed so as to divide the air space into two equal parts, the test results were 14.7 per cent higher than the calculated results. This was due to the infiltration of air through the insulating material used. For wall 53D with an impervious insulating material dividing the air space, the calculated results check with the test results. These results show the effect of excessive air infiltration between air spaces.

When the insulating materials of walls 53B and 53D were cut off 1 in. at the top and bottom allowing the air to circulate around the insulation, the test showed for wall 53B an increase of 33 per cent, and for wall 53E an increase of 53 per cent over the calculated results for the same walls with the insulation sealed at the top and bottom. These tests indicate very definitely that the installation of insulating materials is a very important factor. For commercial insulating materials which are used to divide the air space between the studding, the most important factor to be considered is the sealing of these materials at the top and bottom of the air space in order to prevent a direct transfer of heat around the material by air currents.

The results of this investigation show conclusively that reliable over-all coefficients may be obtained by the hot-box method of testing or by calculations, providing the proper coefficients are available. In practice the variation to be expected in the construction of walls will give a much wider range of difference between calculated results and actual results than has been shown by these tests. This is to be expected in any type of construction, but a large part of the variation may be eliminated by reasonable care in construction and application of the insulating materials.

DISCUSSION

H. S. Ashenhurst (Written): There is one portion of this report which should be given very careful consideration by the members of this Society. I refer to the following extract:

> When the insulation materials of walls 53B and 53D were cut off 1 in. at the top and bottom, allowing the air to circulate around the insulation, the test showed for wall 53B an increase of 33 per cent and for wall 53D an increase of 53 per cent

over the calculated results for the same walls with insulation sealed at the top and bottom. These tests indicate very definitely that the installation of insulating materials is a very important factor.

For the past five years I have devoted practically all of my time to matters connected with insulation of homes. In this work I have had the privilege of examining the installation of different types of insulating materials both in side walls and ceilings. The lack of care used in installing these materials, in a great many cases (possibly for reasons of economy), is startling. In many ceilings examined, a very excellent insulating material has been used, having a low conductivity, but the examination showed that on account of the lack of cleats to hold the material in place, both edges of the insulation where it was supposed to come in contact with the joists, were turned up, leaving an air space of varying thickness. In other cases where the space between the joists was narrower than the width of the insulation, the insulating material had been put in in the shape of a bow, leaving large gaps at the ends for the escape of heat. All these faulty installations of course disregard the manufacturer's specifications, but nevertheless the condition exists today in many homes all over the country.

In side wall construction the same faulty installation exists. A very excellent insulating material is placed at the job where the installation is to be made, but no instructions are given as to where or how it should be installed. The workman may be a very conscientious mechanic, but it is not probable that he has given the proper installation of insulating materials much study. The result is that many cracks are left open for the escape of heat. It is very rare to find a case where close attention has been paid to the sealing of the spaces at the top and bottom of stud spaces, and as shown in this report, this alone means a great reduction in efficiency. The stud spaces act as flues to carry the heat escaping from the walls to the roof and out through the attic space into the air.

This Society could do no greater work for the small home owner than to make it its business to see that when money is spent for insulation, the installation shall be made so as to secure maximum results. I should like to see a committee appointed whose purpose would be the working out of a strict code for the proper installation of all types of insulating materials, and the broadcasting in cooperation with the manufacturers of such materials, the necessity for such proper installation.

C. K. Swift (Written): It is evident that Professor Rowley has settled definitely the long disputed controversy as to the accuracy of calculated overall transmission coefficients. With correct conductivity values as a basis, and proper methods of calculation there should be no uncertainty about the accuracy of computed values.

Unfortunately, however, the value of this work will be largely academic until such a time as we can have available more accurate data on the conductivity values of the numerous building materials in common use. I am particularly interested in, and familiar with fibrous insulating materials, and am greatly surprised to note that Professor Rowley has found conductivity values for boards of this class which are considerably different from the generally accepted values.

Another thing that impresses me in dealing with transmission coefficients is the uncertainty of the third decimal place. Individual conductivity values are never determined with precision to the third place, and when the uncertainty of

the necessary assumptions with regard to mean temperatures is considered, the absurdity of expressing results in this way must be apparent to all. I think it is high time for the engineer to follow the lead of the physicist and express his results in significant figures only.

The magnitude of the error due to air infiltration as shown between walls 53A and 53B is surprising. I take it that in this case only a very moderate air velocity was used. If an error of 14.7 per cent was found with an air velocity of the order of 0.6 mph the error with a 15-mph wind would indeed be serious.

F. C. HOUGHTEN: This paper fills a very important gap in our understanding of heat transmission through building construction. With the many types of building materials on the market and the possibilities of arranging them in different order, we have an almost infinite number of types of construction, and it is impossible practically to determine the overall coefficient for all of them. The heat transmission engineer has developed formulae for determining the over-all coefficients, provided the surface coefficients, air spaces, and conductivities of materials, are known. I think the heat transmission engineer has had sufficient confidence in the application of the formula, but sometimes the heating industry has lacked confidence in its application. Lack of confidence in engineering data is about as serious as erroneous data.

The data contained in this paper tend to clear up that lack of confidence in the application of the formulae for calculating over-all coefficients from the conductivities and surface transmission coefficients.

A. P. KRATZ: I think one of the most important points Professor Rowley has brought out in his paper is the large deviation that can occur when the walls are improperly constructed. It simply points to the fact that the engineer's calculations can be brought almost to naught by improper construction, and more attention has to be paid to both proper construction and proper supervision in the construction. Also, finally, something will have to be done in order to evaluate the walls as they are actually installed. Some of the work we have done in the past winter has indicated that possibly the inside surface temperature may be used as a criterion for evaluating them.

We find that our inside surface temperature is not materially affected by wind conditions, but it is affected largely by the difference in temperature between the indoors and outdoors, and for quite a wide range of wind conditions we get very smooth curves for both the inside and outside surface temperatures. This leads us to hope that it is possible that this means may be used for evaluating the walls as they are actually installed; by getting a comparison of the actual, measured, inside surface temperature with the calculated surface temperatures that should maintain for the given difference in temperature for the wall as computed for the wall properly installed.

F. B. ROWLEY: The differences in conductivity values noted by C. K. Swift are due to the fact that some of these materials were used in the walls at lower mean temperatures than those at which hot plate tests are usually made. His remarks about reporting results to the third decimal place do not apply to a research project, the prime object of which is to determine the accuracy of a particular method of procedure.

Professor Kratz's method of checking the conductivity of walls by means of the surface coefficients should prove very useful.

No. 841

HEAT AND AIR VOLUME OUTPUT OF UNIT HEATERS

By L. S. O'BANNON,[1] LEXINGTON, KY.

MEMBER

THIS paper describes a series of sixteen tests made on three unit heaters for the purpose of supplying information to the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS' Committee on Standard Code for Testing and Rating Steam Unit Heaters. The work is the result of a co-operative agreement between the University of Kentucky and the Research Laboratory of the Society.

The tests were supervised by the author of this paper in co-operation with his assistants and the Director of the Research Laboratory. A full measure of credit is due the men who constituted the crew of testers and who performed well all the duties of their office. These men were R. W. Bozeman, J. C. Lindley, L. L. Massie, J. W. May and E. D. Moore, all of whom were members of the 1929 graduating class of the University of Kentucky.

In testing unit heaters the information usually desired is heat output, air volume output and final air temperature for several fan speeds and for various degrees of entering air temperature.

There is only one generally approved method of measuring heat output. This method consists of operating the unit for a short period of time, usually about one hour, and measuring the quantity of steam condensed; the pounds of steam condensed per hour multiplied by the heat given up per pound of steam gives the heat output of the unit in Btu per hour.

There are two methods in common use for measuring the air volume output. One consists of measuring the air directly by means of some type of flow meter, usually the Pitot tube. The other is called the condensate-temperature rise method. The latter is based on the assumption that the heat absorbed by the air is equal to the heat given up by the steam. The fundamental formula for this method is

$$W_c\ (H-q) = W_a C\ (T_F - T_E)$$

Where W_c=weight of condensed steam in pounds per hour

H=total heat content of steam entering the heater, Btu per lb

q=heat in the condensate leaving the heater, Btu per lb

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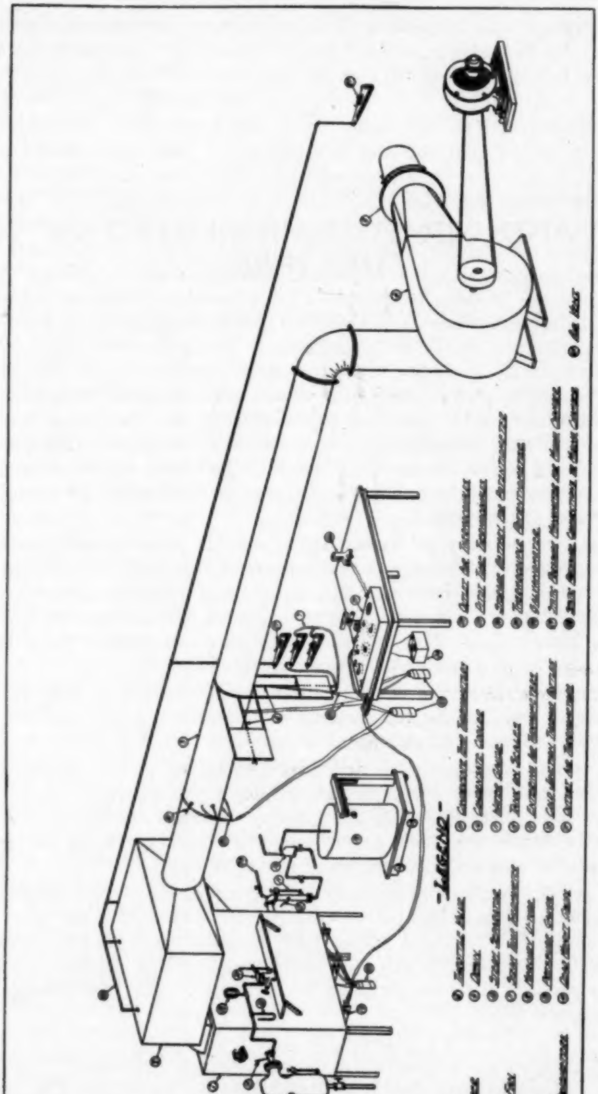

FIG. 1.—DIAGRAM OF A FLOOR TYPE UNIT HEATER SHOWING TEST EQUIPMENT

W_a=weight of air passing through the heater in pounds per hour
C=specific heat of air in Btu per lb per F (usually assumed as approximately 0.24)
T_F=final temperature of the air leaving the heater, F
T_E=temperature of air entering the heater, F

The volume of the air for any unit of time and for any specified pressure and

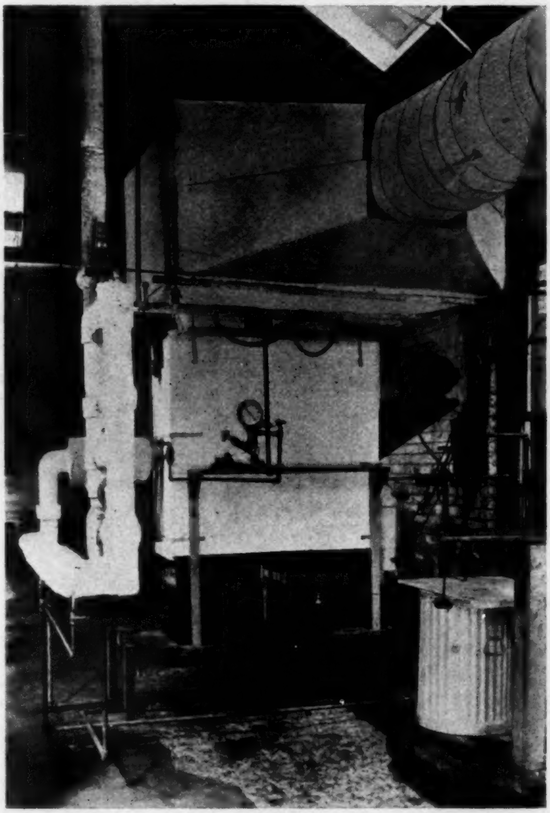

Fig. 2—View of Unit Heater and Mixing Chamber

temperature is calculated from the weight of the air, W_a, which is obtained from the formula after substituting numerically the other values derived from the tests.

W_c is obtained directly by weighing the condensed steam. By referring to steam tables, H is obtained from the steam temperature and pressure observed during the test, and likewise, q is obtained from the temperature of the con-

densate. These values are easily and accurately obtained by ordinary methods. T_s is the average of the readings of several thermometers or thermocouples arranged around the inlet to the heater. As a rule no difficulty is encountered in measuring the entering air temperature. The principal precaution required is to shield the thermometers from the direct radiation from the heater. T_F is more difficult to determine accurately due to the non-uniform temperature and velocity of the warm air usually encountered at the outlet or outlets of the unit. To eliminate temperature indications which do not represent a true average a large duct is provided into which the unit heater discharges. The enlarged duct serves as a mixing chamber and as a collecting chamber also in the case

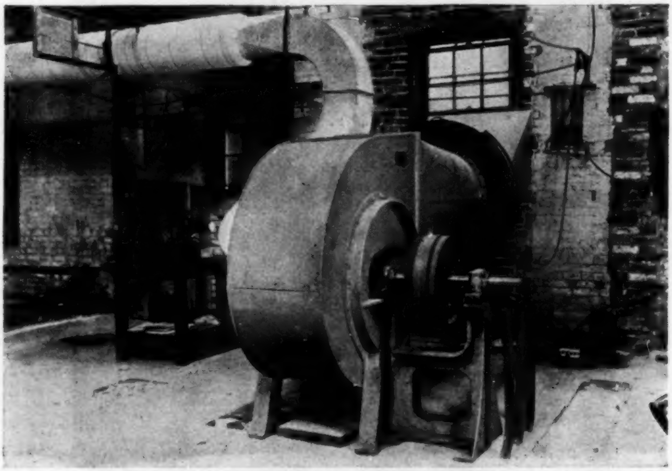

FIG. 3.—VIEW SHOWING PITOT TUBE ASSEMBLY, POTENTIOMETER EQUIPMENT AND EQUALIZING FAN

of a unit with multiple outlets. The air leaves the mixing chamber through a single opening of restricted area at which point the final temperature of the air is measured. The mixing chamber and unit heater are insulated to prevent excessive loss of heat and consequent lowering of the final air temperature.

To overcome the resistance of the mixing chamber an auxiliary fan is required. The fan is connected to the outlet of the mixing chamber and the air flow is regulated by means of a damper so that a static pressure of zero is maintained within the mixing chamber. With this arrangement it is supposed that the unit is operating under the same conditions of free delivery as would obtain if no duct work were attached to the unit heater outlet.

With the Pitot tube method of finding air volumes it is still desirable to know the final air temperature. The same scheme of discharging into a mixing chamber is used and the Pitot tube measurements are made in the duct between the mixing chamber and the equalizing fan.

Fig. 1 shows a diagram of a floor type of unit heater with test equipment

arranged to conform to the methods of testing just outlined. The diagram illustrates the set-up used in this investigation. Several photographic views are shown in Figs. 2-5.

In a code for testing it is desirable to have a single method of procedure. Where two methods are equally in favor the question naturally arises as to which is the more accurate. Not having a third method of absolute accuracy it

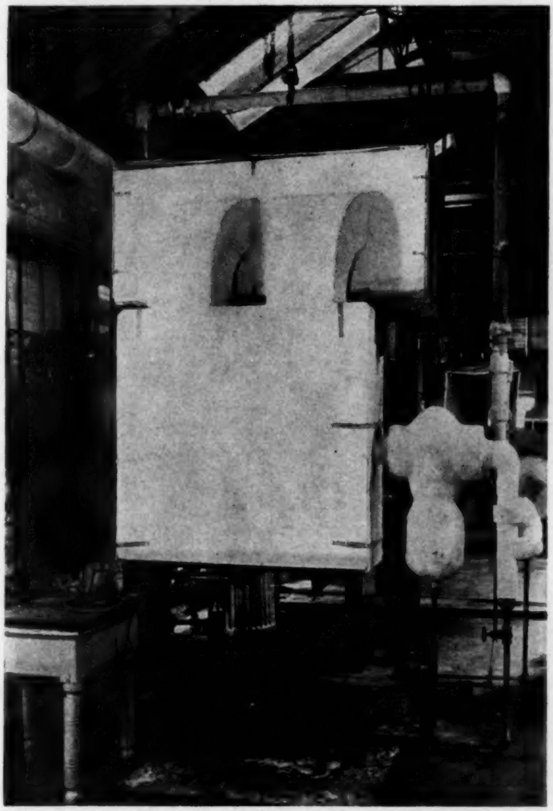

Fig. 4.—Rear View of Unit Heater and Mixing Chamber

is impossible to conclude directly from experiment as to the accuracy of the two methods in question. However, the two methods may be compared one with the other; advantages and disadvantages of each weighed, and the probable errors evaluated.

This investigation was undertaken from the latter standpoint. Nothing unusual or ultra-precise was attempted. Reasonable effort was made to eliminate sources of error and a few obvious corrections were estimated and applied.

The arrangement of the set-up was planned and the application of instruments was executed with no greater elaboration of detail than ordinarily would have been observed in any test of similar nature.

The results of the tests have been summarized in Table 1. It will be necessary to explain some of the items of this summary in detail, following the order

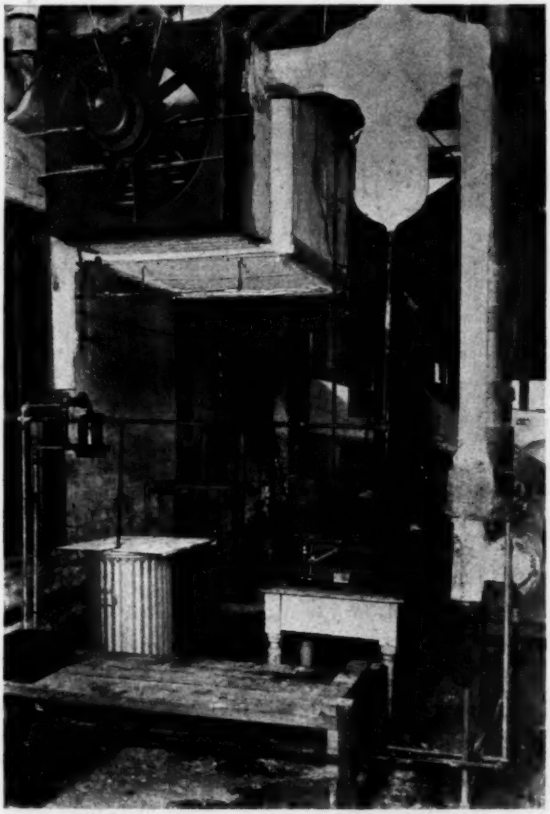

FIG. 5.—VIEW SHOWING CEILING TYPE OF UNIT MOUNTED FOR TEST

in which they are listed, and keeping in mind the chief purpose of the investigation, that is, a comparison of the Pitot tube method with the condensate-temperature rise method of measuring air volume output. Tests numbered 1 to 6 inclusive were all made on the same unit heater which was of the draw-through type with centrifugal fan. Test numbers 11 to 16 inclusive were made on another unit of the same type but of different manufacture. Each of these units had two outlets. Tests numbers 7 to 10 inclusive were made on a blow-through heater with disc fan and a single outlet.

The duration of each test was more or less arbitrary. When five or six readings at 10 minute intervals gave repeatedly practically the same values nothing was to be gained by continuing the tests for a longer period.

A constant steam pressure was maintained to give a saturation temperature of 227 F, which corresponds to 5 lb gage pressure at standard barometric pressure. The steam was supplied from a high pressure boiler and was reduced to the required pressure through a throttle valve operated manually. The steam was maintained at a few degrees superheat to eliminate possible error due to moisture in the steam. During some of the tests it was necessary to inject a slight spray of water into the steam to limit the degree of superheat.

At the outlet of the heater the drawing, Fig. 1, shows a vertical pipe with a *Y* fitting and a water gage. The condensate thermometer was placed in a well screwed into the *Y*. The level of the condensate was held a few inches above the thermometer bulb. A stem correction was applied to this thermometer. To insure elimination of air a vent valve was attached to the outlet of the heater and a thread of steam was allowed to flow through the valve continually. The heat given up per pound of steam was calculated as the difference between the heat content of the steam at inlet conditions minus the heat of the liquid at the temperature indicated by the condensate thermometer.

The condensate was weighed on platform scales capable of being read to ¼ of a pound.

Both thermometers and thermocouples were used to measure entering and final air temperatures. Six thermometers, and six thermocouples connected in parallel, were arranged around the inlet to the heater. Four thermometers, and four thermocouples in parallel, were used at the outlet of the mixing chamber. No correction of any kind was applied to the inlet temperature readings. The inlet thermometers and thermocouples were shielded from the heater but they were not shielded from surfaces in the room (the floor for instance) which may have been at a lower temperature than the entering air.

The final air temperatures, by thermometers, recorded in the summary include a correction for emergent stem and also a positive correction for heat loss by conduction through the insulation on the mixing chamber and the unit heater casing. The emergent stem correction in every test was approximately 0.2 F and was positive in sign. Variations from this value for individual tests were in hundredths of a degree only. The correction for heat losses amounted to from 0.5 F to 2.5 F, depending upon the final temperature of the air. The correction was estimated by merely calculating the transmission loss through the insulation using the temperature rise of the air as the temperature difference and neglecting the influence of air velocity on the inside of the heater and mixing chamber. This correction was also applied to the final air temperatures obtained from the thermocouple readings.

The mean specific heat of the air was calculated from the following formulae

For dry air,

$$C_p = 0.24112 + 0.000009\, t$$

For water vapor,

$$C_p = 0.4423 + 0.00018\, t$$

The specific heat values obtained from a combination of these formulae take into consideration not only the moisture content of the air but also the varia-

TABLE 1.—SUMMARY OF TESTS ON UNIT HEATER

1	Test Number	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
2	Duration of Test, Minutes	50	40	50	50	50	50	40	40	40	40	50	40	40	40	40	40
3	Fan Speed, Rev per Min	1788	1181	792	1778	1165	797	870	870	870	870	834	436	647	841	647	437
4	Steam Pressure, lb per Sq In	5.4	5.4	5.4	5.6	5.6	5.6	5.5	5.5	5.5	5.5	5.4	5.4	5.4	5.4	5.4	5.4
5	Steam Temperature, F	233	230	232	234	236	231	233	236	236	246	238	232	238	232	230	230
6	Heat Content of Steam, Btu per lb	1160.3	1158.3	1159.8	1160.8	1161.8	1159.5	1160.3	1161.8	1161.8	1166.6	1162.7	1159.8	1162.7	1159.8	1158.8	1158.8
7	Condensate Temperature, F	225	225	225	224	224	224	224	224	224	224	224	225	225	225	225	226
8	Heat of Liquid at Condensate Temperature, Btu	193.1	193.1	193.1	192.1	192.1	192.1	192.1	192.1	192.1	192.1	192.1	193.1	193.1	193.1	193.1	194.1
9	Heat Given up by Steam, Btu per lb	967.2	965.7	966.7	968.7	969.7	967.4	968.2	969.7	969.7	974.5	970.6	966.7	969.6	966.7	965.7	964.7
10	Weight of Condensate, lb per Hour	204.3	146.6	105.6	212.1	150.3	111.6	135.8	135.0	97.5	157.1	192.9	116.6	159.4	198.4	162.8	120.0
	Temperature of Entering Air, F																
11	Thermometer No. 1	78.8	81.3	83.5	70.7	73.3	73.3	76.1	76.1	77.7	75.3	70.1	71.2	70.4	69.2	69.0	69.7
12	Thermometer No. 2	78.9	81.8	84.2	71.4	73.5	74.1	77.2	77.0	78.6	76.5	70.3	71.5	70.0	68.6	68.5	69.0
13	Thermometer No. 3	78.1	80.5	82.5	71.0	72.9	72.8	83.6	83.2	83.6	81.4	70.2	71.5	70.1	69.2	68.9	69.7
14	Thermometer No. 4	78.1	80.5	83.0	70.1	72.4	72.9	82.7	82.0	83.2	80.3	70.9	71.8	70.7	69.9	69.7	69.6
15	Thermometer No. 5	78.0	80.5	82.6	70.5	73.0	73.3	78.1	78.4	81.0	77.7	70.9	71.7	70.5	69.6	69.1	70.0
16	Thermometer No. 6	77.7	80.5	82.6	70.4	72.6	72.7	76.7	76.8	79.1	75.8	71.3	71.7	70.5	69.8	69.5	69.2
	Average Temp of Entering Air—																
17	Thermometers	78.3	80.8	83.1	70.7	73.0	73.2	79.1	78.8	80.5	77.8	70.6	71.6	70.4	69.4	69.1	69.5
18	Thermocouples	78.8	81.6	84.2	71.5	73.7	74.1	79.2	79.6	80.7	78.5	71.1	72.2	70.9	70.1	69.6	70.3
	Final Temp of Air, F (including Stem Correction and Temperature Drop Due to Heat Losses)—																
19	Thermometer No. 1	139.9	149.3	157.1	135.6	143.7	151.5	146.7	144.8	144.5	140.7	147.4	162.7	153.5	147.3	153.0	162.5
20	Thermometer No. 2	140.4	149.9	157.8	136.2	144.4	151.9	144.6	147.0	150.5	142.1	148.1	163.8	154.4	148.0	153.9	163.5
21	Thermometer No. 3	138.9	148.0	156.0	134.6	142.6	149.9	145.4	145.9	146.5	141.3	147.8	163.5	154.0	147.9	153.8	162.5
22	Thermometer No. 4	139.1	148.2	155.5	134.9	142.7	150.2	145.9	146.5	153.7	139.4	147.7	163.6	153.9	147.9	153.8	162.9
	Average Final Temp of Air—																
23	Thermometers	139.6	148.9	156.6	135.3	143.4	150.9	145.7	146.1	148.8	140.9	147.8	163.4	154.0	147.8	153.6	162.9
24	Thermocouples	140.1	148.8	156.9	135.6	143.9	151.4	144.7	145.2	149.1	140.3	147.1	162.8	153.0	146.8	154.0	162.5
	Temperature Rise, F																
25	By Thermometers	61.3	68.1	73.5	64.6	70.4	77.7	66.6	67.2	68.3	63.1	77.2	91.8	83.6	78.4	84.5	93.4
26	By Thermocouples	61.3	67.2	72.7	64.1	70.2	77.3	65.5	65.6	68.4	61.8	76.0	90.6	82.1	76.7	84.4	92.2
27	Psychrometer Reading—Dry-Bulb	......	......	......	......	......	......	77.5	77.5	77.5	77.0	72.0	72.0	71.0	71.0	71.0	71.0
28	—Wet-Bulb	......	......	......	......	......	......	66.0	66.0	66.0	66.5	58.5	58.5	58.5	58.5	58.0	58.0
29	—Relative Humidity, %	......	......	......	......	......	......	55.0	55.0	55.0	58.0	44.0	44.0	47.0	47.0	45.0	45.0
30	Mean Specific Heat including Moisture, Btu/lb/F	0.2421	0.2422	0.2422	0.2420	0.2421	0.2421	0.2445	0.2445	0.2440	0.2440	0.2437	0.2438	0.2438	0.2438	0.2437	0.2437

	Weight of Air, lb per Hour—																
31	By Thermometer	13314.	8583.	5734.	13143.	8551.	5739.	8074.	7968.	5659.	9919.	9952.	5036.	7853.	10034.	7634.	5085.
32	By Thermocouples	13314.	8698.	5798.	13245.	8576.	5769.	8210.	8162.	5651.	10127.	10109.	5103.	7722.	10257.	7644.	5152.
	Average of Pitot Tube Readings—																
33	Static Pressure, in. H_2O	−0.711	−0.297	−0.130	−0.698	−0.298	−0.133	−0.317	−0.312	−0.053	−0.580	−0.916	−0.240	−0.530	−0.924	−0.539	−0.290
34	Velocity Pressure, in. H_2O	0.643	0.267	0.112	0.638	0.263	0.113	0.236	0.233	0.113	0.368	0.371	0.093	0.209	0.378	0.217	0.094
35	Barometer Reading, Inches of Mercury	29.15	29.15	29.15	28.82	28.82	28.82	28.88	28.88	28.87	28.87	29.07	29.09	29.09	29.10	29.11	29.12
36	Absolute Pressure of Air at Pitot Tube, in. Hg	29.10	29.13	29.14	28.77	28.80	28.81	28.86	28.86	28.87	28.83	29.00	29.07	29.05	29.03	29.07	29.10
37	Temperature of Air at Pitot Tube, F by Thermocouples	138.2	147.0	153.7	134.4	142.0	147.6	143.0	143.1	146.5	138.7	145.9	159.3	151.2	145.9	151.0	159.0
38	Density of Air at Pitot Tube, lb per Cu ft	.06457	.06370	.06302	.06424	.06350	.06293	.06310	.06309	.06276	.06346	.06325	.06203	.06279	.06288	.06286	.06211
39	Velocity of Air at Pitot Tube, ft per Min	3459.	2244.	1461.	3455.	2231.	1469.	2120.	2107.	1471.	2640.	2655.	1342.	2000.	2688.	2037.	1349.
40	Area of Duct at Pitot Tube, Sq ft.	1.01714	1.01714	1.01714	1.01714	1.01714	1.01714	1.01714	1.01714	1.01714	1.01714	1.01714	1.01714	1.01714	1.01714	1.01714	1.01714
41	Cubic Feet of Air per Min at Pitot Tube, cfm	3518.	2282.	1486.	3514.	2269.	1494.	2156.	2143.	1496.	2685.	2701.	1365.	2034.	2734.	2072.	1372.
42	Static Pressure in Mixing Chamber, in. H_2O	0	0	0	0	0	0	0	0	+0.1	−0.1	0	0	0	0	0	0
43																	
44	Total Heat Output, Btu per Hour	197599.	141572.	102084.	205461.	145746.	107962.	131482.	130910.	94546.	153094.	187229.	112717.	154554.	191793.	157216.	115764.
45	Cfm at 70 F and 29.92 in. Hg, by Pitot Tube	3031.	1940.	1250.	3012.	1923.	1255.	1815.	1804.	1253.	2274.	2280.	1130.	1704.	2294.	1738.	1137.
	Ditto: Condensate—Temp Rise Method—																
46	By Thermometers	2961.	1909.	1275.	2923.	1902.	1276.	1796.	1772.	1259.	2206.	2213.	1120.	1687.	2232.	1698.	1131.
47	By Thermocouples	2961.	1935.	1290.	2946.	1907.	1283.	1826.	1815.	1257.	2252.	2248.	1135.	1717.	2281.	1700.	1146.
	Percentage Condensate Method is High—																
48	Using Thermometer	......	......	2.00	......	......	1.67	......	......	0.48	......	......	......	......	......	......	......
49	Using Thermocouples	......	......	3.20	......	......	2.23	0.61	0.61	0.32	......	......	0.44	0.76	......	......	0.79
	Percentage Condensate Method is Low—																
50	Using Thermometer	2.31	1.60	......	2.96	1.09	......	1.05	2.38	......	2.99	2.94	0.88	1.00	2.70	2.30	0.53
51	Using Thermocouples	2.31	0.26	......	2.19	0.83	......	......	......	......	0.97	1.40	......	......	0.57	2.18	......
52																	
53																	
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tion of specific heat with temperature. No psychrometer readings were taken during the first six tests and the air was assumed to be dry.

The weight of the air, items 31 and 32, was calculated from the fundamental formula for the condensate-temperature rise method already explained.

The Pitot tube was used according to standard practice. The duct was divided into five areas requiring 20 readings to complete a traverse. Only one traverse was made during a test. Two Pitot tubes were used, one for the horizontal traverse and one for the vertical traverse. Inclined tube draft gages were used for measuring pressures. The gages were checked for accuracy by a comparison with a sensitive manometer patterned in principle after a micromanometer described by John L. Hodgson in the April, 192?, issue of *Instruments.*

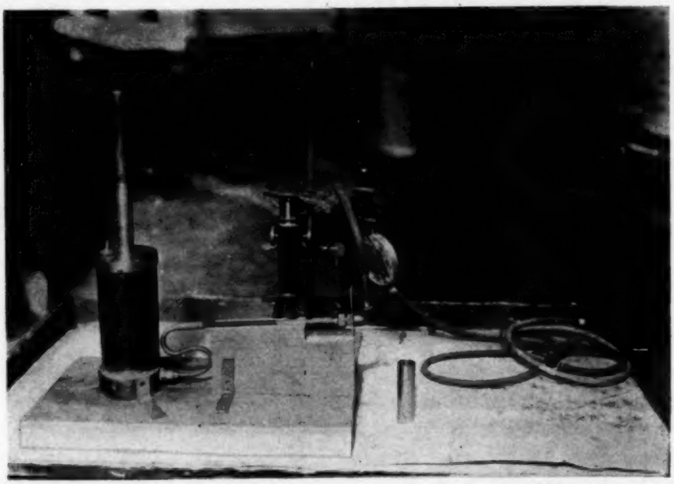

FIG. 6.—MANOMETER USED FOR CALIBRATING DRAFT GAGES

The manometer is shown in Fig. 6. The design of the manometer is such that the pressure readings are proportional to the vertical displacement of a plunger in a chamber of oil. Table 2 gives calibration data for the draft gage used for measuring velocity pressures.

To convert Pitot tube readings into air velocities the density of the air at the Pitot tube must be known. This means that the temperature of the air must be known. Temperature readings were made simultaneously with Pitot tube readings by means of a thermocouple fastened to the side of the Pitot tube. The average temperature thus obtained was used in computing the air densities. Excepting tests numbers 1 to 6 the vapor content of the air was also considered. In Fig. 7 are typical curves showing how the temperature and velocity pressure varied across the duct.

The air volume output by both methods has been reduced to a standard density of 0.0749 lb per cu ft. This density corresponds to dry air at 70 F and 29.92 in. heating pressure.

Assuming the Pitot tube method as a standard, percentages have been calculated showing the amount the condensate method is high or low. Glancing at these figures it is quite apparent that the two methods agree fairly well. In fact the agreement is so close that the writer finds himself in the paradoxical situation of having to explain a fact which from a rational viewpoint is axiomatic and yet so contrary to the experience of a majority as to be seriously questioned. The writer is referring to the almost universal conception that volumes obtained by the Pitot tube method are consistently much higher than volumes obtained indirectly by means of the condensate method. The thought is sometimes expressed by saying that the results by the condensate method are conservative.

If an immediate conclusion of practical value is to be drawn it is sufficient to say that these tests show that for any engineering purpose the difference between the air volume outputs obtained by the two methods is negligible. This statement, however, does not tacitly condone slipshod practices.

Table 2.—Calibration Data for the Draft Gage Used for Measuring Velocity Pressure in Unit Heater Tests

Ellison Gage Reading Inches of Water	Standard Gage Reading Inches of Water	Error
0.130	0.1314	—0.0014
0.146	0.1458	+0.0002
0.183	0.1810	+0.0020
0.186	0.1866	—0.0006
0.198	0.2000	—0.0020
0.224	0.2210	+0.0030
0.349	0.3522	—0.0032
0.408	0.4066	+0.0014
0.471	0.4737	—0.0027
0.507	0.5081	—0.0011
0.554	0.5527	+0.0013

Returning to the percentages of error, the average amount by which the condensate method was low, using thermometers, was 1.3 per cent; using thermocouples, 0.11 per cent. The percentages range from — 2.00 per cent to +2.99 per cent, using thermometers; and from — 3.20 per cent to + 2.31 per cent, using thermocouples. The larger negative values were obtained at low air velocities and the higher positive errors were obtained at the higher velocities. At first thought this would seem to indicate a systematic error in using the Pitot tube, especially in measuring velocity pressures, since the possibility for error becomes greater at low velocities. On the other hand, the final air temperatures increase as the air velocity decreases. It is quite probable that the temperature readings are in error for the same reason that one may suspect error in the Pitot tube. This is especially true with respect to the corrections which were added to final temperatures, since as was previously pointed out, these corrections were estimated on the basis of still air conditions on both sides of the insulation.

The variation of the percentage differences with respect to air velocity would have disappeared if the corrections applied to final temperatures had been a little greater at the low velocities and a little less at the high velocities.

There are many factors involved. For instance, it is stated in paragraph 83, Appendix 1, of the CODE OF MINIMUM REQUIREMENTS FOR THE HEATING AND VENTILATION OF BUILDINGS, that the specific heat of air "varies from 0.2375 to 0.2430 as determined by various investigators." These two values differ by approximately 2.3 per cent, which is more than enough to materially change the character of the results of the tests under consideration.

In regard to the measurement of air temperatures, thermocouples are to be preferred to thermometers on account of thermocouples being less influenced by radiation and air velocity effects, and also on account of their being adaptable to unusual circumstances.

The Pitot tube method has been considered the standard for determining air

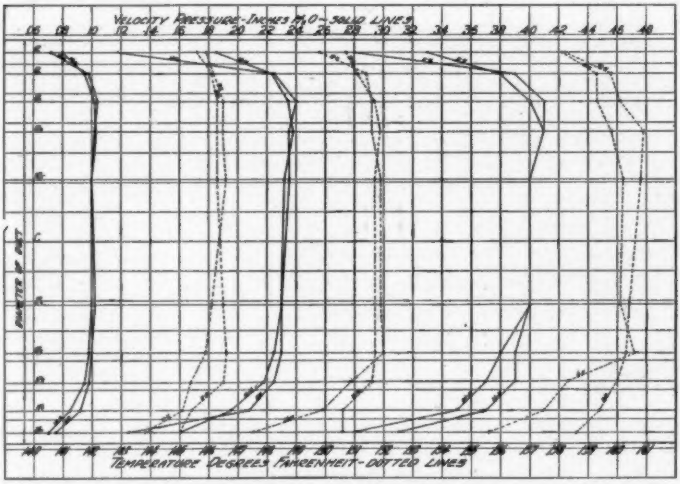

FIG. 7—CURVES SHOWING VARIATION OF TEMPERATURE AND VELOCITY PRESSURES ACROSS DUCT

volumes. The Pitot tube is fundamentally accurate in theory and when properly designed it has proved accurate in practice. It would seem logical, therefore, to include the use of this instrument in a code for testing unit heaters. But since the testing of unit heaters includes other observations from which it is possible to calculate the áir volume output with sufficient accuracy, it seems unnecessary and superfluous to include the pitot tube or any other flow meter device such as the orifice or venturi tube.

The condensate method requires fewer instruments, less labor in testing and in making calculations, and a smaller amount of floor space for the set-up.

DISCUSSION

PROF. A. P. KRATZ: We had occasion to test a unit heater, and we made a traverse with a thermocouple on a Pitot tube in a similar way to Mr. O'Bannon's work here, except that we used the full face of the heater and divided it

up into squares. When we used the average square root of the head and the average temperature rise calculated from the thermocouple readings in each individual square, we found that we had more heat put into the air than was given up by the steam, and finally we had to divide the face into more squares. After using about 32 squares, we tried the method of regarding each square as a unit and calculating the heat that was passed through that unit from the individual thermocouple reading and the individual velocity head at the center of that square, and then summed up all the heats of the individual squares to obtain the total heat. I would like to ask whether Professor O'Bannon tried that scheme, and if he did, whether it modified his results, whether it made them worse or better. We found in our case that we got much better correlation. The correlation, of course, would depend on how uniform the temperature was, and how uniform the velocity distribution was. The better the temperature distribution, and the better the velocity distribution, the more nearly would the summation equal the results as calculated from the average square roots of the heads and average temperatures.

The second point—we have had occasion to measure the pressure loss through our warm air furnace systems. We found that in order to do so we had to use the Wahlen gage. Our total pressure losses through the warm air furnace system were in the order of thousandths of an inch of water rather than hundredths of an inch of water. I think the maximum change from the total capacity of the furnace down to practically no capacity was a little greater than 0.01 in. of water. I would like to ask Professor O'Bannon whether he had any trouble maintaining a pressure near enough atmospheric so that it did not affect the flow. A very small difference in pressure there would affect the gravity head. That is one of the points that I see that I think Professor O'Bannon will have to be very careful of.

D. E. French: I would like to answer Professor O'Bannon's direct question about the agreement between the measurement of air quantity, on unit heater test, between the Pitot tube method and that of condensation temperature rise. I believe there is no rational basis for assuming that there should be any disagreement. I think it is rather complimentary to the accuracy of the method that Professor O'Bannon has used and of the observation and of the corrections not ordinarily made, that he should get that agreement. I think the fact that previous observers have been unable to get an agreement so close, has been due entirely to their use of the instruments involved and to their failure to make corrections which must be estimated under difficulty without very elaborate test set-up and instruments.

H. S. Wheller: I would like to ask whether Professor O'Bannon has in his investigation formed any conclusion as to whether or not accurate tests could be made by any method other than the plenum chamber method that he has used. There seems to be some opinion that heaters can be tested accurately by other means and it is possible that Professor O'Bannon's work may have indicated something in that direction.

Homer R. Linn: I am not interested in unit heaters, but a few weeks ago I was called in as an arbitrator on a heating matter. I found that their main point of discussion was unit heaters. They had a unit heater installed very similar to the one in Fig. 5, discharging the air between 1500 and 1800 ft velocity. Their complaint was that they were condensing too much steam with

cold flow. They could not keep the workmen in this garage. The man had another garage which was similar in construction to this one, larger, and when I figured the heat loss of both buildings, it had a greater heat loss but was condensing less steam. It had a unit heater—I do not see one like it in here—it was set on the floor, about 18 in. off the floor, taking the air from the bottom of the heater and discharging that at a velocity of about 550 ft per minute, condensing much less steam than the one of high velocity.

I wish in their code or in these tests they would in some way or other tell us how we could test those two, other than putting them in a plenum chamber, which does not mean anything to the fellow who is paying for the coal.

L. A. HARDING: I think the principal differences in test results are the items mentioned by Professor Kratz. The difference lies there. Any difference beyond that point is perhaps due to errors in the steam tables or the specific heat of air assumed. There should, of course, be no difference between the two methods if we assume the steam tables are correct and specific heat of air employed is correct.

W. A. ROWE: It might be well to acquaint the Society with one or two points with reference to this code that would perhaps cast some light on it, that has to do with the history to this date. This committee is composed of men who are, on one hand, in the fan business, accustomed to testing air, and other manufacturers who are in the heater business alone. In the early stages of the meeting of the committee there seemed to be a clearage between the method of determining the air capacity; those who had been testing fans rather preferring a method of measuring the air direct, and those not in that business, but in the heater business, preferring to calculate the air capacity by measuring the temperature rise. They were both agreed as to the proper method of getting the Btu rating from a condensation test, and about the one point which seemed to prevent the adoption of the more simple method of getting the volume of air from the temperature rise, was that there was a prevailing opinion among the best engineers we could find that any calculated volume of air on a temperature rise basis would show less air than on the Pitot tube test, for the reason that the temperature rise as read on the thermometer would be affected by radiant heat and would have to be reduced before you calculate your volume. The consensus of opinion from those who had tested heaters was that that might be as much as 3 or 4 per cent. It seemed unwise to incorporate in a code a method for which there did not seem to be a rational basis for saying that there should be such a difference, or secondly, to incorporate in a code an arbitrary percentage for which you had no backing. Therefore, we asked the Research Laboratory in conjunction with the University of Kentucky, who did the work under Professor O'Bannon, to tell us whether there should be any difference, and secondly, if there was, how much.

I think it has been quite clearly indicated from the tests they have already made, that you are perfectly safe for all practical reasons in using whichever method is more convenient. There should be close agreement. We are indebted to them, I think, a great deal for that. With that background then of these tests, we are perfectly justified in drawing up a code which will represent the easiest, most practical means which will give that degree of accuracy. The committee adopted the temperature rise method to calculate the air volume, feeling that that was the easiest way to do it.

There were just one or two points, however, that I think prevented a definite recommendation to adopt this code at this time, and that is this. While all the testimony that has been presented to date shows the consistency of results achieved through the collecting chamber set-up, it has not been proved that those results are consistently correct or incorrect as compared to the performance when the heater is disconnected from the chamber and operated on the floor. In other words, the heater behaves in a certain way on the floor, but if you put it on a collecting chamber that chamber may effect the volume of air passing through it. In order to make certain that with the chamber set-up and with a zero static on the chamber there was no disadvantageous effect on the performance of the unit, it was determined to use a floor test on the unit for condensation. Then connect it to the chamber and run the chamber test as shown in this code at zero static, and check the condensation thus obtained. When they were brought into agreement it was reasonable to expect that the device for measuring the temperature had no effect on the heater capacity.

Bear in mind that we are trying with that chamber so far to get an accurate temperature reading. To do that we collect the air and measure it at a reasonably high velocity after it is collected and mixed, and these results have proved undoubtedly that you can make readings that way and actually get the correct temperature rise agreeing with the Pitot tube air volume determination for that condition of test. It seemed, however, that it might be possible to obtain a temperature rise equally satisfactorily without all that set-up, and from now on until the next meeting we have an opportunity for the members of the group to make their floor test and chamber test afterwards for determination of the temperature rise. If it is found that with the floor tests and possibly some other simpler means, thermocouples or resistance wires, etc., they can get substantially the same temperature readings on a floor test as they do with this chamber, it would be possible to still further simplify this practice by doing away with the chamber set-up.

That covers some of the points that I thought might be put into the picture to give a little better understanding of what we have tried to accomplish and the situation of the testing practice in this code to date.

Prof. L. S. O'Bannon: I made my original presentation as short as possible with the anticipation that what was lacking in it would come out in the discussion, and Mr. Rowe and Mr. French have supplemented my remarks very well. Professor Kratz' remarks really hit at the heart of this investigation so far as the tests are concerned. He was getting down to the details of measuring heat and temperatures. The only method we used for measuring the temperatures, was simply by means of stationary thermometers or thermocouples at the outlet of the mixing chamber, and that was for the purpose of getting the final temperature leaving the heater, and then we had thermocouples on the Pitot tubes and temperature reading taken at the same time as the traverse was made for the purpose of getting the temperature of the air at the Pitot tube. Now, the manometers we used were ordinary inclined tube draft gages, and the pressures we were reading were in the order of hundredths of an inch of water rather than thousandths of an inch, and we presume that the gages we used were accurate enough.

Mr. Harding mentioned variations which might affect the results, such as

variations of specific heat and values of properties of steam—you will notice in the table of results we have percentage variations as far as 3 per cent on one side through zero to minus 3 per cent on the other side, and I consider that those errors represent experimental errors which it was impossible to eliminate and also take care of recognized values and specific heat and properties of steam, etc. That kind of error I was not especially interested in when I put the question with regard to the difference between the Pitot tube and condensate-temperature-rise method. It is the general impression I have gotten in discussing this with various people that there seemed to be some fundamental difference between Pitot tube results and condensate-temperature-rise results, which would lead you to expect that you could never have a set-up that would make the two agree, in spite of the fact that we would consider elementary errors of experimental work, and variations of specific heat, properties of air and steam, which have been mentioned. The whole secret of this set-up, as far as accuracy is concerned, is the thorough insulation of the mixing chamber in the duct where you are taking these temperatures. With thorough insulation of this chamber you get very slight variation of temperature across the duct, so that the numerous temperature readings mentioned by Professor Kratz are not necessary, because the temperatures are fairly uniform.

Mr. Rowe has explained the reason for this mixing chamber. As far as this comparison of Pitot tube method and condensate method is concerned, the results are not affected by the characteristics of this mixing chamber with regard to whether or not the heater is reproducing its present delivery when it discharges into the mixing chamber.

So far as I know, no other method has been proposed which could be used as a general method for testing all types of heaters. Every method I know of uses some kind of a chamber on either the discharge side or the inlet side, usually the discharge side for measuring this final temperature. If final temperature is not important, you can merely set up a duct for measuring air velocities, providing some means for reproducing free delivery of the heater. But this entire practice, as far as I know, hinges around the use of this mixing chamber.

The quesiton as to whether this mixing chamber reproduces present delivery conditions to the heater, is a question which certainly needs investigation, but that was not considered as a vital point in this investigation.

MR. FRENCH: It just occurs to me that the gentleman who referred to field testing of unit heaters was not answered. I would just like to say that the Code Committee, in its deliberations so far, has considered first, as being of the greater importance, the accurate laboratory methods by which manufacturers could test and rate unit heaters, but which of necessity require laboratory methods and laboratory set-ups. The Code Committee has been asked to provide a code for the field testing of unit heaters. That is the next labor of the Code Committee. Of necessity a field test cannot be surrounded with the instruments and devices that are possible in a laboratory. The same degree of accuracy cannot be obtained, but a code will be drafted for field tests, and a corresponding tolerance will be set for the results expected with the less elaborate equipment adapted for that use.

No. 842

ERRORS IN THE MEASUREMENT OF THE TEMPERATURE OF FLUE GASES[1]

By P. NICHOLLS[2] (*Member*) AND W. E. RICE[3] (*Non-Member*), PITTSBURGH, PA.

THE determination of the temperature of the flue gas is an essential operation in all tests of boilers, and it is important that the measurements should be accurate. Errors may arise from a number of causes which can be broadly divided into: (1) Errors due to inaccuracies of the thermometer or instruments used with it, (2) errors due to the failure of the thermometer to indicate the true temperature of the gas stream in contact with it, and (3) errors due to the fact that the temperature as measured may not represent the true average of the whole gas stream.

Errors due to inaccuracies of the temperature measuring instruments used will not be discussed. It is common practice, and it is also most convenient, to use thermocouples in conjunction with a millivoltmeter or a potentiometer—the latter by preference. With reasonably cheap instruments the error can be kept within plus or minus 5 deg.[4] Too much dependence should not be placed on one couple; it is advisable to have two holes in the flue pipe, one for the couple being used and the other for the insertion of another couple to ascertain whether the first has maintained its calibration. The permanent insertion of two couples, connected to the instrument by a double-throw switch, is a safe practice, since it rarely happens that both couples go bad at the same time.

The object of this paper is to discuss the second source of errors—namely, those due to the failure of the couple to register the true temperature of the gas stream in contact with it. Statements were made during the January, 1927, meeting of the Society, at conferences and in the discussions of the Codes for the Testing and Rating of Low Pressure Boilers, that the flue gas temperature measured in the way required by the Code for Testing might differ as much as 200 to 250 deg from the true temperature of the gas because of the exposure of the thermocouple to surfaces at temperatures different to those of the gas. The committee which prepared those codes had recognized that there would be some error due to this cause and had pointed this out in an earlier report; and,

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[4] The Fahrenheit scale is used in this paper.

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furthermore, they had suggested that the instructions for testing were rigid enough to insure that the error would be approximately the same for boilers of the same output.

Fig. 1 will be used to illustrate the principles underlying the measurement of temperature by an exposed thermocouple. A couple in the center of the flue at *A* will take heat from the gases flowing by because of its contact with them; but, because it is exposed to the metal of the boiler through the spherical angle or cone *aAá*, it will radiate heat to these colder surfaces. The surfaces of the flue pipe will also be at a lower temperature than the gases, although this difference will be reduced by the insulation of the flue; the couple will, therefore, be radiating heat to the flue also. As the couple is gaining heat by conduction

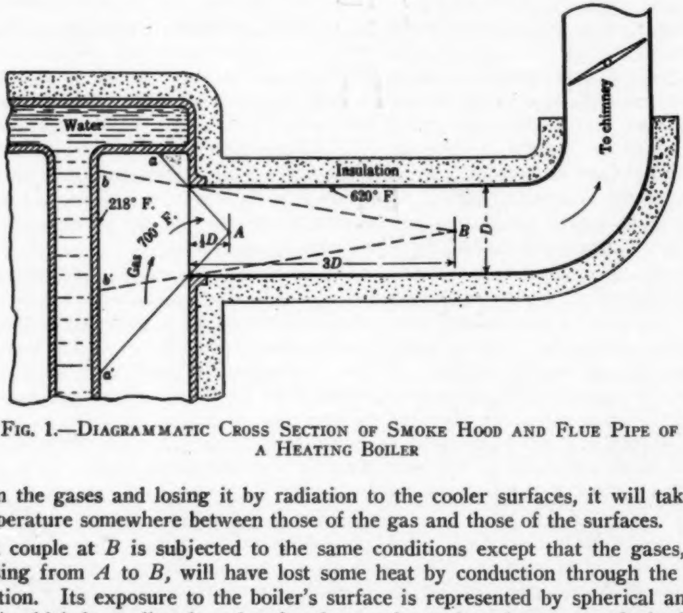

FIG. 1.—DIAGRAMMATIC CROSS SECTION OF SMOKE HOOD AND FLUE PIPE OF A HEATING BOILER

from the gases and losing it by radiation to the cooler surfaces, it will take a temperature somewhere between those of the gas and those of the surfaces.

A couple at *B* is subjected to the same conditions except that the gases, in passing from *A* to *B*, will have lost some heat by conduction through the insulation. Its exposure to the boiler's surface is represented by spherical angle *bBb'*, which is smaller than that for the couple at *A*, and consequently its loss of heat by radiation to the boiler surfaces will be less than that of *A*, although this decrease is counteracted somewhat by a large exposure to the surface of the flue.

The temperature registered by either couple will, therefore, be lower than the true temperature of the gas; with a well-insulated pipe, unless its diameter is very small, one would expect the temperature of *B* to be greater than that of *A*, and that *B* would more nearly indicate the true temperature of the gas.

It will be noted that the relative loss of heat by radiation of couples at *A* and *B* is independent of the diameter of the flue provided the distances of *A* and *B* from the end of the pipe are the same proportions of the pipe diameter, ½*D* and

3D in the figure. The relative temperatures registered by A and B will therefore also be independent of the diameter of the pipe except for the effect on the gas temperature of the loss of heat through the insulation; the proportionate amount of heat lost this way for a fixed velocity of gas in the flue will increase with decrease in pipe diameter; that is, the gases will have a larger drop in temperature as they travel along the flue as its diameter decreases. This loss of heat can be computed with fair accuracy and it is therefore not necessary to experiment with a range of boiler sizes and pipe diameters; safe deductions on the relation between the observed and the true temperature might be made by tests on a few sizes.

There is, however, one factor which will cause a thermocouple in the center of a large-diameter flue to depart less from the true temperature of the gas in contact with it than one in a small flue: Heat radiations pass through air without being absorbed, but carbon dioxide and water vapor have an appreciable absorptive power, so that they will shield the couple; this shielding will increase with the thickness of the layer of gas—that is, with the radius of the flue.

Tests were made on four existing set-ups of boilers. No attempt was made

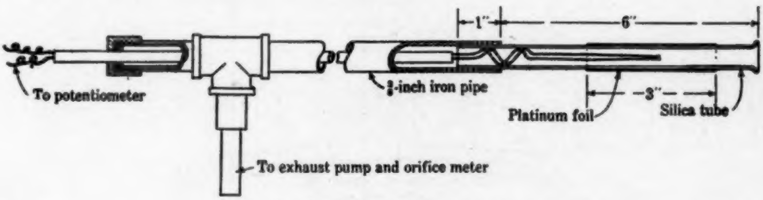

FIG. 2.—SHIELDED THERMOCOUPLE

to obtain scientific data to connect the results of the tests with computations; the work was confined to measurements under conditions conforming to boiler test practice. Several thermocouples were placed along the flue, their junctions in all cases being on its center line; an additional couple was peened into the metal of the flue to register its temperature. The thermocouples were all of No. 22 gage (0.0253 in.) and were calibrated before the tests.

The true temperature of the gas was measured at one place by what is usually termed the velocity method. Fig. 2 shows the device used; its principle consists in the shielding of the couple and the drawing of the gases over it at such a high velocity that the loss of heat by radiation is small compared with the gain by conduction from the fast-moving gas; in addition the couple is shielded from direct radiation to the cold surfaces by the silica tube which, further to reduce radiation, had a bright platinum foil wrapped around it. The radiation coefficient of the platinum to iron surface would be about 0.06 as compared with silica to iron surface of 15 times as much. The junction of the couples and a length of wire from it were kept central and free from contact with the silica tube by forming a spiral of the wires as shown in the figure. Porcelain supports in the silica tube would be more convenient.

Fig. 3 shows the first set-up tested; it consisted of a 5-in.-diameter vertical flue pipe attached to a vertical sectional boiler. The positions of the exposed

couples are indicated by the numbers. The shielded velocity couple is designated as *Vc*, and the surface couple as *Sc*. Observations were made at rates of burning giving flue gas temperatures of approximately 400, 700, 800, 900 and 1,000 deg. One of the couples was attached to a recording potentiometer so that the temperature of the flue gas could be followed; when the temperature had remained constant at a desired value for at least 10 min, several series of readings of the couples were taken and the values for each couple were averaged

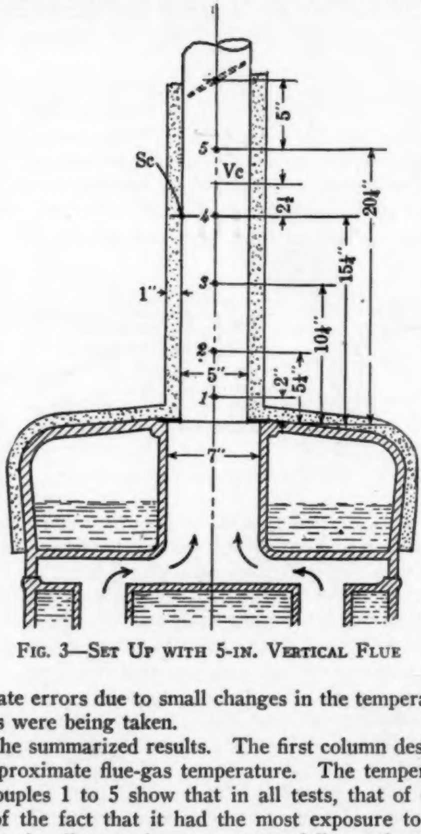

FIG. 3—SET UP WITH 5-IN. VERTICAL FLUE

in order to eliminate errors due to small changes in the temperature of the gases while the readings were being taken.

Table 1 shows the summarized results. The first column designates the series of tests by the approximate flue-gas temperature. The temperatures registered by the exposed couples 1 to 5 show that in all tests, that of couple 1 was the highest in spite of the fact that it had the most exposure to the cold surface of the boiler; also, in all tests the temperature falls as the distance from the boiler increases. This fall in temperature registered by the exposed couples is greater that can be accounted for by the cooling of the gases because of loss of heat to the surface of the flue; for the 800 deg test, the difference between the temperatures of couples 1 and 5 was 21 F; the computed drop in temperature due to loss of heat through the insulation is 9 F. It is probable that this lack of agreement is partly due to the center of the gas column as it enters the flue

Table 1.—Comparison of Temperatures, F, Measured by Exposed and Shielded Thermocouples in Set-up Shown in Fig. 3.

Approximate Temperature	Exposed Thermocouples					Shielded Thermocouple Between 3 and 4	Metal Surface	Approximate Error of Exposed Thermocouple
	1	2	3	4	5			
400	405	404	397	393	393	400	353	5
700	717	714	710	698	696	714	588	10
800	830	826	816	819	809	827	713	10
900	915	907	899	899	890	909	787	10
1000	1002	996	989	986	979	1000	875	12

being at a higher temperature than the gas at the circumference which has been in contact with the surfaces of the boiler.

The last column shows the error in the temperatures registered by the exposed couple; these values are the difference between the true temperature and that of an exposed couple located at the same position as the shielded couple.

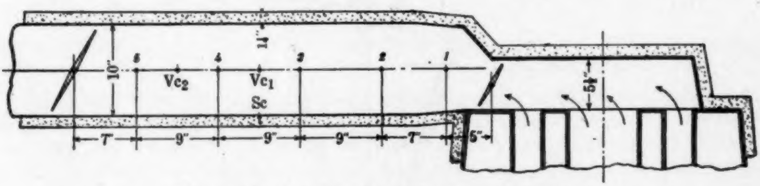

Fig. 4.—Set Up with 10-in. Horizontal Flue

The error has an approximately constant value of 10 F over the range of 700 to 1,000 F.

Fig. 4 shows another set-up that was tested. The diameter of the flue pipe in the set-up of Fig. 2 was small for the boiler used, whereas in Fig. 4 it was rather large. Also, the flue pipe was horizontal, had little exposure to cold boiler surfaces, and had a damper which, though wide open, would tend to disturb the flow of the gases.

The tests were conducted in the same manner as previously and Table 2 shows the results.

The values show the same general relations to each other as those in Table 1 but the error of the exposed couple is larger, being about 30 F. Some of this

Table 2.—Comparison of Temperatures, F, Measured by Exposed and Shielded Thermocouples in Set-up Shown in Fig. 4

Approximate Temperature	Exposed Thermocouples					Shielded Thermocouple Between 3 and 4	Metal Surface	Approximate Error of Exposed Thermocouple
	1	2	3	4	5			
400	420	418	414	408	408	422	340	10
700	724	719	715	707	706	737	608	26
800	833	822	817	804	803	843	705	33
900	919	916	913	903	902	938	813	30
1000	1013	1010	1009	998	...	1034	910	30

increase may be due to the lower velocity of the flue gas which, for similar temperatures, would be under two-thirds of that in the previous tests; also traverses of the flue diameters showed larger variations in temperature and a greater drop as the sides were approached, which would be expected with the horizontal pipe and the damper disturbing the flow.

As a check on the tests with the 10-in. flue the position of the shielded couple

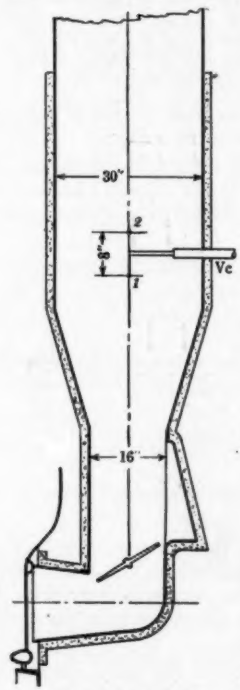

FIG. 5.—SET UP WITH 30-IN. VERTICAL FLUE

was changed from between couples 3 and 4 to between 4 and 5. Table 3 shows the results of these tests.

The results are very similar to those of Table 2. There was an error of only 5 F at the 400-deg temperature, as against an error of 10 F in the previous test. This is explained by the higher temperature of the metal pipe, the difference between the true temperature and the metal temperature being 35 deg in Table 3 and 82 deg in Table 2, so that the exposed couple lost less heat by radiation and would consequently register a temperature nearer to the true value.

As a further check on the values with a 10-inch diameter pipe, the set-up of Fig. 4 was varied by using a short horizontal flue and then a vertical leg. Three exposed thermocouples were placed in the vertical flue, 14, 23, and 32

TABLE 3.—COMPARISON OF TEMPERATURES, F, MEASURED BY EXPOSED AND SHIELDED THERMOCOUPLES IN SET-UP SHOWN IN FIG. 4

Approximate Temperature	Exposed Thermocouples					Shielded Thermocouple Between 4 and 5	Metal Surface	Approximate Error of Exposed Thermocouple
	1	2	3	4	5			
400	405	401	398	397	394	400	365	5
700	743	737	729	718	717	750	596	32
1000	1001	998	995	988	985	1020	868	34

TABLE 4.—COMPARISONS OF TEMPERATURES, F, MEASURED BY EXPOSED AND SHIELDED THERMOCOUPLES IN A 10-IN. DIAMETER VERTICAL FLUE

Approximate Temperature	Exposed Thermocouples			Shielded Thermocouple Between 2 and 3	Metal Surface	Approximate Error of Exposed Thermocouple
	1	2	3			
400	395	392	391	397	364	5
700	692	690	691	706	614	16
800	793	790	789	805	720	15
900	902	897	895	917	807	21
1000	990	986	990	1014	884	26

in. from the center line of the horizontal flue, with the shielded couple midway between the last two. Table 4 shows the results obtained.

The errors shown by the last column are of the same order as those with the horizontal flue, but somewhat smaller. It will be noted that in the 800-deg tests the difference between the readings of couples 1 and 3, which were 18 in. apart, was only 4 deg, whereas the difference was 15 deg for the same distance in the horizontal flue. The difference in temperature computed from the heat lost through the insulation is about 4½ deg; this shows that the gases must be stratified as they leave the boiler but are well mixed by the right-angle bend.

Fig. 5 shows the set-up used in the test of a 30-in. diameter vertical flue.

TABLE 5.—COMPARISONS OF TEMPERATURES, F, MEASURED BY EXPOSED AND SHIELDED THERMOCOUPLES IN SET-UP SHOWN IN FIG. 5

Approximate Temperature	Exposed Thermocouples		Shielded Thermocouple Between 1 and 2	Metal Surface	Approximate Error of Exposed Thermocouple
	1	2			
400	395	388	402	348	10
500	475	470	482	406	10
600	596	593	607	520	13
700	712	711	725	576	16

TABLE 6.—TEMPERATURES, F, MEASURED BY EXPOSED THERMOCOUPLES IN SET-UP SHOWN IN FIG. 6

Approximate Temperature	Exposed Thermocouples			
	1	2	3	4
400	405	413	408	394
700	706	690	686	682
800	875	865	861	849
900	941	937	931	915
1000	1033	1034	1026	1015

The couples had no exposure to the boiler surfaces and only two were used, with the shielded couple midway between them; they were located about 2½ pipe diameters from the start of the flue. The flue was insulated one pipe diameter beyond the last couple. Table 5 shows the results of the test.

The computed average drop in temperature along the flue caused by loss of heat through the insulation would be small, although the insulation was only 1 in. thick; in the test at 600 deg it would be about half a degree in passing

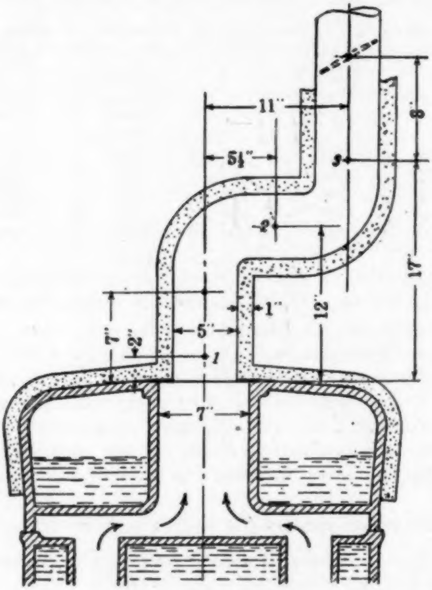

FIG. 6.—SET UP WITH 5-IN. VERTICAL FLUE WITH BENDS

from couple 1 to couple 2. The differences registered became less as the temperature increased, that is, as the velocity of the gas was greater, because there would be better mixing.

The exposed couples gave temperatures 10 to 16 deg lower than the true gas temperatures and showed greater uniformity of difference than in the tests of the small-diameter flues. Traverses across the flue diameter showed that the temperature did not, on the average, vary more than 2 or 3 deg within 5 in. on each side of the center of the pipe.

The occasion was also taken to make some measurements in the set-up which was available as shown in Fig. 6. The object of this was to see whether the couple located at position 3 would show higher values than the couple at position 1. Couple 1 is exposed to the cold surfaces of the boiler, whereas couple 3 is entirely shielded from them. No shielded couple was used in these tests.

Table 6 shows the results. At 400 deg couple 3 was higher than couple 1 but at all other temperatures it was lower. It would, therefore, not seem to be of any advantage to measure the temperature in the bend.

SUMMARY AND CONCLUSIONS

The measurements here reported were all made with an insulated flue pipe; larger departures from the true temperature would have been found with an uninsulated pipe.

In all the set-ups tested the temperature registered by the thermocouple nearest the boiler was higher than those farther away, showing that the stratification of the gases has more influence than has the exposure of the couples to the cooler surfaces.

The temperature registered by the exposed couple in the 5-in. and 30-in. diameter flues departed from the true gas temperature by amounts of the same order—10 to 15 deg. The differences in the 10-in. diameter horizontal flue were greater, but no proved explanation of this fact can be offered; that it was due to differences in the radiation loss is not probable because the corresponding surface temperatures of the metal in Tables 1 and 2 are approximately the same. The more reasonable explanation is that the stratification of the gases was such that the velocity couple did not draw into its tube a stream of gas representative of that flowing by the exposed couple. There is a possibility, however, that the temperatures as given by the shielded couple in the 5-in. diameter flue are low because the size of the pipe was too small to properly insert the shielded couple device.

The tests on the 30-in. flue give the most reliable data on the departure of the exposed couple from the true temperature of the gas in contact with it, as the shielded couple had a stream of gas at constant temperature to draw from.

Although a thermocouple in the center of the flue indicates a temperature lower than the true temperature of the gas in contact with it, yet this lower value will not differ much from the true average temperature of the gas in the flue because the temperature of the gas in the center of the flue is higher than the average over the cross section.

The tests here reported have therefore shown that the use of an exposed thermocouple installed as specified by the Boiler Testing Code of the Society and with the flue insulated will indicate a temperature which will approach that of the true average temperature of the gases. The probable error should be within 10 deg.

It is suggested that a standardization of the position of the thermocouple for test purposes should be with its junction on the center line of the flue. The best location would be at 4 equivalent flue diameters from the boiler because this would give time for the gases to mix; this might not always be convenient and it would be satisfactory to allow not less than 2 or more than 4 diameters. The size of the thermocouple wire should not be more than No. 20 *B* and *S* gage (0.0319 in.).

DISCUSSION

R. V. FROST: I would like to make one comment on this paper. We have heard a great deal about the inability to measure the stack temperatures ac-

curately, and here are the results of reading temperatures of five thermocouples at different points in the outlet to the boiler. It is interesting to see how closely these agree. The greatest error or the largest difference in any case does not exceed five per cent. Now, if we can read our stack temperatures within five per cent, we cannot call it a great error. The paper also shows that in taking temperatures of 1000 F, the maximum difference is a trifle over one per cent. The paper serves to demonstrate that there is not such a great error in reading stack temperatures as we sometimes think.

No. 843

DETERMINING THE QUANTITY OF DUST IN AIR BY IMPINGEMENT

The results of cooperative research work between the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the University of Minnesota

By F. B. ROWLEY[1] (*Member*) AND JOHN BEAL[2] (*Non-Member*)
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ONE of the major problems confronting the Technical Advisory Committee on Air Filters has been that of determining the quantity of dust in the air. Many methods have been suggested and tried out, a part of which have been more or less successful. While several of the methods have some merit, it has not been a simple problem to select one which has been acceptable to the industry as a whole. For those methods which have been found to give consistent results, it has generally been necessary to employ rather delicate and expensive apparatus which can only be handled by trained operators.

To be acceptable, a method should not require apparatus which is out of the reach of the average engineer. It should be applicable to the average air condition and should give dependable results. To fulfill the latter requirements, it is not necessary to have absolute results, but if the instrument does not record on the absolute basis, the relation of the results obtained to the absolute quantities should be the same for like conditions of air. There are, probably, several methods which may be found to fulfill these requirements, but thus far none have been generally accepted.

It is not necessary that all of the existing methods be approved or disapproved, but rather that some of the most promising ones be investigated with the idea of selecting at least one which may be used as a standard or as a tentative standard in order that results obtained by different engineers may be on a comparable basis.

The first research work conducted was with the AA dust determinator. The results of this work were reported in a paper entitled, A Study of Dust Determinators, presented at the Society's Semi-Annual Meeting, 1928, and pub-

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Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Bigwin Inn, Lake-of-Bays, Ontario, Canada, June, 1929.

lished in the TRANSACTIONS, Vol. 34, 1928, p. 475. These researches indicated that this method has promise and undoubtedly could be refined to a point where consistent and reliable results may be obtained. The recent studies which have been made and the results of which are reported in this paper relate to the impingement type of dust determinator. This method has some advantages, such as simplicity, convenience to operate, and also the possibility that the operator may get a better idea of the quality of dust when examined under the microscope.

In general, the apparatus consists of a nozzle through which the air for test is drawn at high velocity by a pump. The stream of air strikes a plate which is usually covered with some viscous material. The dust particles striking the plate remain and may be examined under a microscope or by other means.

With this type of dust counter, there are several questions which immediately arise:

1. What is the limiting size of the particles which should be considered?
2. What proportion of dust is actually taken from the air?
   This is affected by such factors as:
   A. The distance of the nozzle from the plate.
   B. The velocity at which the air is brought against the plate.
   C. Shape and position of the nozzle.
   D. The density and size of particles of the dust.
3. Is the proportion of dust consistent and uniform for samples of the same air when treated in the same manner?

In the first part of the investigation the Hill dust counter was used, one model of which is illustrated in Fig. 1. This consists of a nozzle *A*, a glass plate *B* and a pump *C*. The nozzle is set at a definite distance from the plate and the air is drawn through it by a plunger pump. The samples of dust taken by this method are counted under the microscope, which in this case is an integral part of the instrument. The microscope furnished with the apparatus magnifies to 80 diameters. After some experience with the instrument, it was found that the operator could count with a reasonable degree of certainty the particles visible under the microscope. There was, however, some question as to the lower limit of dust particles to consider. When these same samples were put under a high power microscope, only a small portion of the sample could be observed at one time and there was also present a much greater number of extremely small particles which would make the process of counting impossible unless these particles were eliminated. There are two plausible reasons suggested why these should be eliminated:

First. That the number of these particles is proportional to the number of larger particles so that a count of the most distinguishable ones furnishes a relatively correct count for the whole sample.

Second. The size of these particles may be below the size which is of any interest in air filtration work.

In considering these two reasons it is difficult to find a basis supporting the first suggestion. On the other hand, the second seems to be a necessary conclusion. If the dust particles decrease in size indefinitely, there must be some lower limit beyond which it is impractical to go. This viewpoint might be considered to be supported by the work of Whitlaw Gray and Speakman (*Proceed-*

ings, Royal Society, A102, 615-27, 1923), who determined the concentration, average size, and average mass of particles of zinc oxide suspended in air. They determined the number of particles by direct observation with an ultra-microscope, and the mass by filtering the air and weighing the filtrate on a micro-balance. Their results show that in a diluted cloud of zinc oxide smoke there are in the neighborhood of 1,000,000 particles per cu cm, or about 3,000,000,000 particles per cu ft. The possibility of having such a high concentration of extremely minute particles seems to indicate that either these small particles must be disregarded in air filtration work or that the idea of separating the particles by impingement and counting them must be given up.

In the preliminary tests carried out by the Hill dust counter, it was found that by counting only those particles which were large enough to appear under the microscope as a definite black point, reasonable check results could be obtained. The best results seemed to be given with 200 to 300 particles collected on the

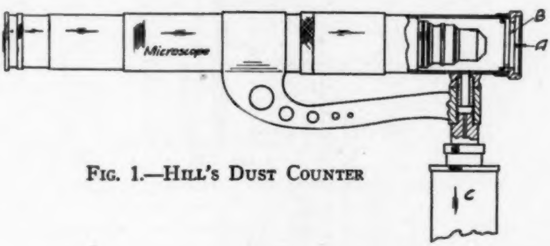


FIG. 1.—HILL'S DUST COUNTER

glass. It was also found that air with low dust content gave the most consistent results. The results obtained by the various type of Hill dust counters will be discussed later in the report.

SPECIAL EQUIPMENT

For the purpose of studying the various factors which affect the proportion of dust taken out of the air by the impingement method, two special pieces of apparatus were constructed. The first, shown by Fig. 2, was a device which caused the air to be passed through three consecutive nozzles and impinge against three consecutive plates. The nozzles are shown by letters, A_1, A_2, and A_3, and the plates, B_1, B_2 and B_3. The air from the last nozzle chamber discharges into a pump chamber. The second piece of apparatus is illustrated by Figs. 3, 4 and 5. Fig. 3 is a diagrammatic view of the apparatus which consists essentially of two vacuum tanks *A*, a water ejector *B* for creating a vacuum, a mercury manometer *C* and a dust sampling chamber *D*. The dust sampling chamber is shown in detail in Figs. 4 and 5: Fig. 4 being vertical, cross-sectional; and Fig. 5 a lower view with the dust plate in place. Referring to Fig. 4, *A* represents an air-tight chamber which is connected to the vacuum line, and *B* is a removable top which contains three nozzles, *C*. These nozzles are closed at their upper end by a spring operated valve *D* and project at their lower ends to a predetermined distance from the glass plate *E*. The glass plate is held in position by springs *F* and serves as a collector for the dust samples.

The distance of the nozzle *C* from the plate *E* may be changed to suit the requirements. The cover *B* is held in place by atmospheric pressure and the smooth joint between *A* and *B* is sealed with vaseline. The object of the apparatus is to obtain a dust sample under conditions of uniform air velocity.

In operation the water ejector is used to pump the air out of the tanks, *A,* the vacuum in these tanks being measured by manometer, *C.* Since the tanks are of known capacity, the amount of air in them at any time may be measured by the pressure in the manometer tube or by closing off the ejector and opening one of the valves, *D,* Fig. 4, the amount of air passing through the nozzle may

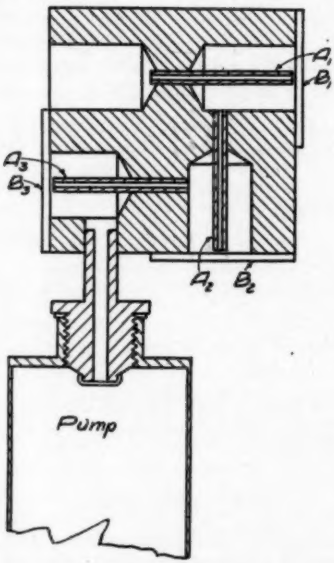


FIG. 2. COUNTER, WITH THREE NOZZLES IN SERIES

be determined by the change in pressure. From this process the velocity through nozzle may be calculated for different pressures and by adjusting the vacuum tanks to these predetermined pressures a dust sample may be taken at any given velocity through the nozzle. The volume of air for a test may be measured either by noting the time during which the valve is open or by calculating the increase of volume of air in the tanks by the manometer readings. This apparatus was used both for the single nozzle and for the nozzles in series as shown in Fig. 2.

TYPES OF NOZZLES

In those tests which were conducted with three nozzles in series, two types of nozzles were used: *First,* those with square ends; *second,* those with sharp edges.

The nozzles were all 0.05 in. in diameter and were set at various distances from the plate. It is evident that a long line of experiments might be conducted covering the size, shape and placement of the nozzle, but it is not necessary to know all of the details of this part of the problem in order to select satisfactory conditions. The size and shape of the nozzle and the velocity of air through it must be governed by the size of the sample required. Any combinations of conditions which can be readily duplicated, giving consistent samples uniformly distributed under the microscope, should be satisfactory.

The results of comparative tests between the square and sharp edged nozzle seemed to indicate that the latter gave the best results. The sharp edged nozzle

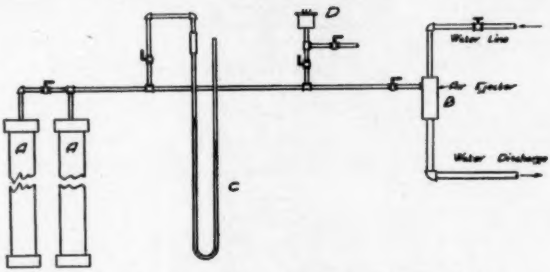


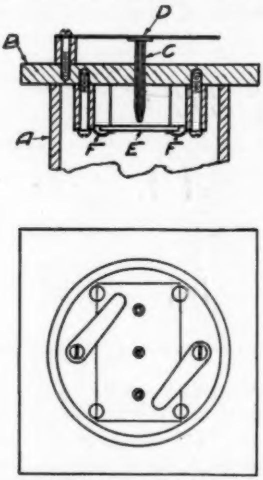
FIG. 3.—CONSTANT VELOCITY DUST DETERMINATOR

was, therefore, used for the remainder of the tests. The first series of tests in which the three nozzles were used in series was made by drawing the air through with a plunger pump. The results of these tests are shown in Table 1. They showed that, with the nozzle set at 0.005 in. from the glass, as high as 94 per cent of the total dust was taken out at the first impingement. When this same air was tested with a single nozzle, a much higher count was obtained. The only reasonable explanation of this seemed to be that, since the same pressure drop was used across the three nozzles as across the single nozzle, the velocity through the three nozzles was necessarily much lower and reduced the amount of dust collected.

TABLE 1.—THREE NOZZLES IN SERIES, DUST SAMPLES TAKEN WITH PLUNGER PUMP

	No. of Samples Average	Clearance Between End of Nozzles and Plates, Inches			Average Dust Part. Per Cu Ft	Maximum Per Cent Variation from Average	Average Percentage Retained on 1st Plate	Maximum Per Cent Variation From Average
		1	2	3				
Square ended nozzles	7	0.005	0.005	0.005	2251	15.5	83.0	14.0
	3	0.010	0.005	0.005	7059	5.0	89.6	1.6
	8	0.016	0.005	0.005	2700	44.4	75.7	18.0
	1	0.025	0.005	0.005	7048		71.3	
1st Nozzle with sharp edged exit	4	0.005	0.005	0.005	2609	8.3	88.6	5.0
	4	0.005	0.005	0.005	6998	3.3	94.6	6.1
	6	0.010	0.005	0.005	2048	11.0	86.8	8.5
	4	0.016	0.005	0.005	1552	9.2	85.6	3.1

As it was impossible to change the velocity conditions with a pump method, the apparatus as described in Figs. 3, 4 and 5 was designed and built. In using the vacuum apparatus, it was desired to get both the amount of air used for a sample and the velocity of impingement. As stated before, velocity might be determined from volume and time or from the pressure across the nozzle and a calibration curve for the nozzle. The time required for a test is so short that it is difficult to take it accurately; therefore, the velocity was determined from the pressure. Since the resistance through the nozzle is greatly increased when



FIGS. 4 AND 5.—VERTICAL CROSS SECTION AND UNDER VIEW OF *D* IN FIG. 3

the nozzle is set close to the plate, the velocity is also greatly decreased. It was, therefore, necessary to make a separate calibration for each setting of the nozzle.

By setting the nozzles at different clearances from the plate and changing the velocity at which the samples were taken, it was found that the number of particles of dust collected increased with the velocity of air through the nozzle and decreased as the clearance increased. The curves of Fig. 6 show the results of a series of tests made to demonstrate these characteristics. Further consideration of the results obtained shows that there is a limit to which the decrease in clearance may be used for increasing the dust count. At 0.005 in. clearance, the resistance to the passage of air was so great that 200 fps was the maximum velocity obtainable at full atmospheric pressure. At greater clearances much higher velocities could be obtained and the additional amount of dust collected in the sample by these higher velocities more than off-set the loss due to the greater clearance between the nozzle and glass. This illustrates the reason why with three nozzles in series, 0.005 in. clearance appears to give the best results. The resistance of the three nozzles in series is so high that the addi-

tional effect of reducing the clearance is comparatively small and a reduced clearance increases the dust collected in a greater portion than it decreases the velocity.

TESTS WITH CONSTANT VELOCITY

The results of a second series of tests, in which the air was drawn through three consecutive nozzles with the constant velocity apparatus, are shown in Table 2. In these tests the velocity through the nozzles was 250 fps and the clearance was 1/16 in. These tests showed that 95 per cent of the total dust collected was retained on the first slide. The consistency of the percentage taken out indicates that it is not necessary to use more than one nozzle.

TABLE 2.—THREE NOZZLES IN SERIES. DUST SAMPLES TAKEN AT CONSTANT VELOCITY OF 250 FPS. NOZZLES SET AT 1/16 IN. CLEARANCE

Test No.	Slide No.	Dust part. Collected	Per Cent of Total Retained On First Slide
1	1st Slide	over 600	
	2nd Slide	15	
	3rd Slide	6	
		Total 621	96.5%
2	1st Slide	over 1000	
	2nd Slide	48	
	3rd Slide	12	
		Total 1060	94.5%
3	1st Slide	over 800	
	2nd Slide	24	
	3rd Slide	3	
		Total 827	96.8%

TABLE 3.—NUMBER OF LARGE PARTICLES OF DUST DEPOSITED AT VARIOUS VELOCITIES

Velocity of Air Through Nozzle	No. of Particles Above 15 Microns in Size	Velocity of Air Through Nozzle	No. of Particles Above 15 Microns in Size
200	49	400	52
220	42	430	60
270	48	450	51
330	49	460	47
390	49		

The influence of velocity on the number of particles collected was next more carefully worked out. All nozzles were set at the same clearance in order that a larger number of samples might be taken in a shorter time, thus partially eliminating the variation in the amount of dust contained in the air for the different samples. With the nozzle clearance of 0.015 in., the results as given in the curves of Fig. 7 were obtained. These curves are located differently due to the fact that the samples were taken on different days, and, therefore, with a different dust content in the air. They do, however, all show similar characteristics—that is, a general increase in the dust count from 200 to 275 fps with a flat portion between the range of 275 to 300 fps from which the slope extends upward very rapidly. The points fall very consistently on the curves until the

portion of rapid increase in count is reached, after which they show a considerable amount of variation. It will be noted that the samples were taken with three different nozzles which were supposed to be identical. Later work showed that there was a slight variation in the bore of one of these nozzles which gave it a slightly different characteristic. The same general characteristic is also shown in the curves of Fig. 8. These curves show that at lower velocities the count is much higher for the close setting of the nozzle tip to the glass, but that the limiting velocities are soon reached and from this point on the count is higher with a free discharge opening.

It was thought possible that the large number of dust particles collected at high velocities was due to the fact that as the velocity is increased many more of the smaller particles are precipitated to the plate. When counting the sample, it is extremely difficult to differentiate between the two sizes and the total number visible is the most practical count to be made for the sample. One series

TABLE 4.—SAMPLES TAKEN WITH VACUUM APPARATUS, CLEARANCE=1/16 IN.

Series	Sample No.	Velocity fps	Dust Count Particles per Cu Ft	Per Cent Variation from Average
A	1	300	23,800	+0.42
	2	300	24,050	+1.48
	3	300	23,200	+1.90
			Average 23,700	
	4	400	69,000	—1.8
	5	400	71,100	+1.2
	6	400	70,700	+0.6
			Average 70,270	
B	1	300	21,200	+7.90
	2	300	18,810	—4.23
	3	300	19,100	—2.80
	4	300	19,500	—0.76
			Average 19,650	
C	1	300	12,480	—2.82
	2	300	12,410	—3.40
	3	300	12,690	—1.18
	4	300	13,200	+2.64
	5	300	13,280	+3.36
	6	300	13,000	+1.18
			Average 12,843	
	7	400	42,800	+1.33
	8	400	42,800	+1.33
	9	400	44,900	+1.56
	10	400	41,200	—0.33
	11	400	39,500	—6.48
			Average 42,240	

of tests was, however, run in which only particles above a certain size were counted. This was accomplished by comparing the particles to a spot in the field of the microscope which was estimated to be about 15 microns in diameter. The velocity was varied through the entire range and the results obtained are shown on Table 3. These results would indicate that the assumption is correct, although no sharply defined relation exists between the velocity and the smallest size deposited, for even at the lowest velocity many of the particles appear to be just as small as those at the higher velocities, although there were a great many more of the small particles at the higher velocities.

IMPORTANT FACTORS

From the experiments thus far conducted, it is evident that the amount of the dust taken out of a sample is affected by at least three factors.

TABLE 4.—SAMPLES TAKEN WITH VACUUM APPARATUS, CLEARANCE=1/16 IN. (*Continued*)

Series	Sample No.	Velocity fps	Dust Count Particles per Cu Ft	Per Cent Variation from Average
D	1	300	9,600	—4.87
	2	300	9,480	+2.58
	3	300	8,520	—6.91
	4	300	9,240	+0.94
	5	300	8,930	—1.36
			Average 9,154	
E	1	300	29,800	—1.7
	2	300	27,100	—7.5
	3	300	30,800	+5.1
	4	300	28,300	—3.4
	5	300	31,100	+6.1
	6	300	28,900	—1.3
			Average 29,300	
	7	400	84,900	+2.16
	8	400	75,600	—9.03
	9	400	82,900	—0.24
	10	400	86,600	+4.22
	11	400	87,000	+4.70
	12	400	81,600	—1.80
			Average 83,100	
F	1	300	3,160	+3.5
	2	300	2,980	—5.4
	3	300	3,250	+4.5
	4	300	3,180	+1.0
	5	300	3,270	+3.8
	6	300	3,200	+1.6
	7	300	3,000	—4.8
			Average 3,150	

A. Size and shape of the nozzle.

B. Distance of nozzle from sample plate.

C. Velocity of air through nozzle.

The size of the nozzle seems to be governed largely by the size of the sample desired. The shape is subject to further experimental work, but of the nozzles thus far tried the sharp edged nozzle appears to give the best results. The effect of distance between the nozzle and the glass plate depends somewhat upon the resistance to the air flow, the effect being greater for small clearances. From the test made, it would appear that 1/16 in. clearance gives reasonable results and it is a clearance from which wide variation need not be expected

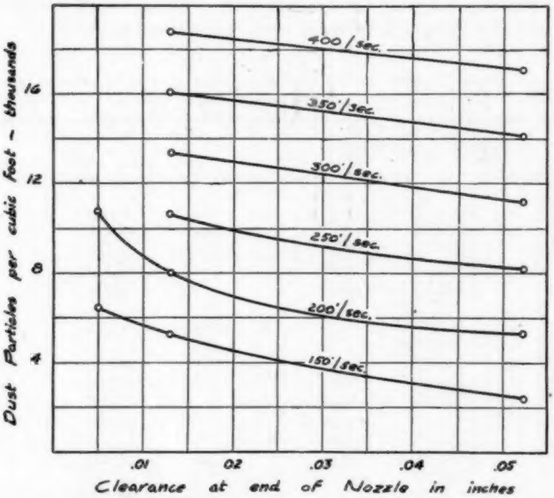


FIG. 6.—EFFECT OF NOZZLE CLEARANCE AND AIR VELOCITY ON DUST COUNT

for small differences either way. This clearance was, therefore, selected for further experimental work. From the curves thus far shown, it is evident that velocity has a very great effect upon the amount of dust deposited and that some specific velocity must be maintained for comparative results. It is also evident that, under no obtainable conditions will all of the dust be precipitated from the sample; therefore, any results with this type of counter must be on a comparative basis. Under this condition those factors which affect a sample must be selected and fixed in some range within which it is possible to remain in practice. The particular values which are assigned to these variable factors do not appear to be of as much consequence as is the necessity for their standardization.

In order to determine the reliability of the impingement method when conditions are definitely controlled, many sets of tests have been run under fixed

conditions. For the greater part of these tests a nozzle 0.05 in. in diameter and with 1/16 in. clearance from the collecting glass was used. Several series of samples were taken from the air in a large room. In each series the conditions of sampling were held constant and the time intervals between the individual samples of a series were made as short as possible in order to eliminate

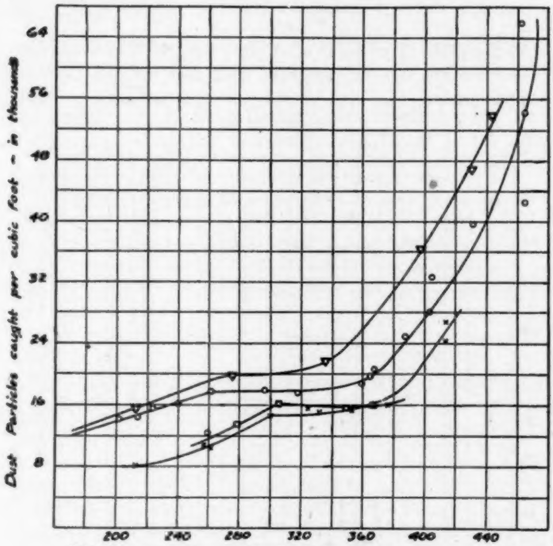


FIG. 7.—EFFECT OF AIR VELOCITY ON DUST COUNT FOR DIFFERENT SAMPLES

variations due to changes in the natural dust content of the air. The velocities used for the different series were from 300 fps to 400 fps. The results for a representative group of these tests are shown in Table 4. They show a maxi-

TABLE 5. RESULTS OF TESTS WITH HILL COUNTERS

Sample No.	Hill Counter No. 1		Hill Counter No. 2		Hill Counter No. 3		Constant Velocity Counter	
	Dust per Cu Ft	Per Cent Variation from Average	Dust per Cu Ft	Per Cent Variation from Average	Dust per Cu Ft	Per Cent Variation from Average	Dust per Cu Ft	Per Cent Variation from Average
1	15,780	— 6.2	14,150	— 12.7	18,080	— 5.3	18,220	
2	13,590	— 19.2	20,950	+ 36.0	13,590	— 28.8	19,490	+ 6.9
3	17,500	+ 4.0	17,500	+ 7.2	14,040	— 26.5	17,280	— 5.2
4	16,340	— 2.9	15,780	— 3.0	13,590	— 28.8	18,280	+ 0.3
5	18,420	+ 9.5	14,400	— 11.2	28,800	+ 50.8	18,140	— 0.4
6	19,320	+ 14.9	14,960	— 7.9	26,500	+ 38.7	17,940	— 1.5
Average	16,820		16,290		19,100		18,225	

mum variation in any case of 9 per cent in either direction from the average. The possibility for a variation from the average would be greater as the number of samples are increased for each set, although from the work done it would seem that six samples are sufficient for reasonable results.

CHECKING RESULTS

As a check on the plunger pump type of dust counter and a comparison of this method with the constant velocity method, dust was sampled by three different types of Hill dust counter and also by the constant velocity method. The Hill

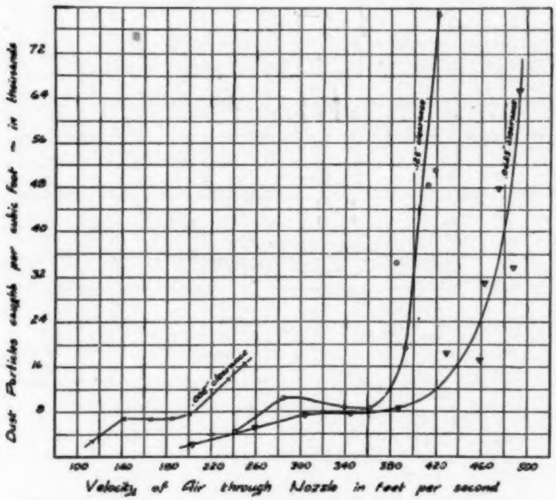


FIG. 8.—EFFECT OF AIR VELOCITIES ON DUST COUNT FOR NOZZLE CLEARANCES

counters used in making these tests are numbered 1, 2 and 3. Counters Nos. 1 and 2 were of the single orifice type with microscope attached. For No. 1 the orifice was in a glass plate at a distance of 0.083 in. from a fixed glass sample plate. No. 2 was of the same type excepting that the orifice was in a metal plate and the sample plate was removable. No. 3 was an older type in which the orifice and pump were separate from the microscope.

The constant velocity apparatus is the one described in Figs. 3, 4 and 5.

The results of these tests are shown in Table 5. An analysis of the results shows that the variation in count for the samples taken at a constant velocity is much less. Since the principal difference in the method used was in the control of velocity through the orifices, it would indicate that this is the factor which might be changed to improve the Hill counter.

CONCLUSIONS

The final conclusions reached in this investigation were:

First: That the amount of dust taken out of the air by the impingement method depends upon several variable factors. Among these factors are size and shape of nozzle, position of nozzle with reference to the collecting plate and velocity of air through nozzle.

Second: The determinations by such a method must necessarily be relative as it is impossible under any conditions to take all dust out of the air and count the same.

Third: The size of dust particles in the air varies through a wide range, and it is evident that some minimum size must be selected below which it is not necessary to count the particles. This lower limit may be set with reference to the quality of the air desired.

Fourth: By holding the conditions constant under which the samples are taken, results may be obtained with the impingement type of counter which are reasonably consistent.

Fifth: There is nothing gained in accuracy by placing nozzles in series.

DISCUSSION

H. H. KIMBALL AND I. F. HAND (WRITTEN): Counts of the number of dust particles per cubic centimeter contained in the atmosphere have been made on the campus of the American University on nearly every working day since December 7, 1922, with the Owens' dust counter, and since September 1, 1928, counts have been made with the Hill dust counter. Observations were also commenced with the Hill dust counter early in September, 1928, at the following additional Weather Bureau stations: Boston, Mass.; New York, N. Y.; Pittsburgh, Pa.; Detroit, Mich.; Chicago, Ill.; Cincinnati, O.; Atlanta, Ga.; Madison, Wis.; Lincoln, Nebr.; Denver, Colo.; and Salt Lake City, Utah. The Hill dust counters were furnished by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. The Society also prepared the instructions for obtaining the dust counts, and towards the end of 1928 sent out a representative to inspect the dust counting, so as to insure uniformity of method at the different stations.

However, the results did not seem to be in accord with known conditions at some of the stations. For example, Pittsburgh showed less dust than any other station, and Cincinnati was a close second; while Madison, Wis., appeared to be one of the smokiest of the twelve cities named above.

Early in May Mr. Hand was sent out equipped with both a Hill and an Owens' dust counter to check up on the measurements at Pittsburgh, Cincinnati, Madison and Chicago. Chicago was included because the count was in question, and because it was necessary to pass through that city to reach Madison.

At Pittsburgh it was found that a lens in the objective of the microscope had become loosened, so that the instrument could not be focused on the dust particles collected. As a result, only a few of the coarsest particles could be indistinctly seen. After properly adjusting the microscope so that it would focus the dust count was increased about ten-fold.

At Cincinnati the observer had interpreted the instructions to mean that only coarse particles were to be counted, and particles that fell on any of the lines of the grid were to be omitted. After including in the count all particles that could without doubt be seen, and excluding only those on or beyond the outside lines of the grid, the number of particles counted at that station was increased about three-fold.

However, two observers almost never obtain precisely the same results with the Hill dust counter when reading simultaneously. Thus, at Pittsburgh, after the microscope had been put in order Mr. Hand obtained with his instrument 219,705 dust particles per cubic foot, while Mr. Zellon, with the Pittsburgh instrument obtained 175,070 particles per cubic foot. At Madison, Mr. Hand obtained 20,496 particles per cubic foot, Mr. Piippo, with the Madison instrument obtained 23,485 particles. At Cincinnati Mr. Hand obtained 44,835 particles per cubic foot, while Mr. Devereaux obtained 51,240 particles.

Repeated comparisons indicated that the discrepancies were at least partly due to uneven dust distribution throughout the atmosphere. A perusal of the paper by Rowley and Beal indicates that other factors, such as quickness of pump stroke, keenness of the eye of the observer, etc., also operate to cause discrepant readings.

The comparisons between dust counts by the Hill and the Owens' instruments are of interest. Individual readings give the following ratios for counts by the Owens' instrument as compared to counts by the Hill instrument:

Date	Place	Ratio $\frac{\text{Owens}}{\text{Hill}}$
May 6	Pittsburgh, Pa.	1564
7	Chicago, Ill. (Univ.)	1953
7	Chicago, Ill. (Federal Bldg.)	1966
8	Madison, Wis.	1646
8	Madison, Wis.	1305
9	Cincinnati, O., (Abbe Obsy)	1547
9	Cincinnati (Federal Bldg.)	1851

The average ratio for Washington for the months September-December, 1928, and February-May, 1929, is 1342, while values on individual days in April, 1929, varied from 5250 to 464.

While both these instruments depend upon the impingement and adhesion of dust particles to a microscope cover glass, they are quite different in principle. The Owens' instrument deposits a coating of water on the particles, thereby not only giving them an adhesive surface, but also increasing their specific gravity. As a result they are deposited on the cover glass in a well-defined line with sharp edges. If quickly removed from the counter the line of water-covered particles is distinctly visible; but the water evaporates, leaving the line invisible before it can be mounted on the microscope slide. When examined under a powerful microscope the edge of the line of dust particles is still sharply defined.

In the Hill counter the dust particles are left dry and light and the receiving surface is covered with an adhesive. There is great scattering of the dust particles, so that only those in a prescribed area are counted, and a factor,

1/0.81, is used to take account of those that fail to adhere or fall outside the prescribed area. Furthermore, the hole admitting the air stream through the cap to the adhesive surface is not well centered, and the way in which the cap is put on makes considerable difference in the proportion of the total dust that falls in the area over which the count is to be made. Also, unless the adhesive is very carefully spread over the receiving surface, particles of it may be mistaken for dust.

It appears to the authors that accuracy has been sacrificed for cheapness in the construction of the Hill dust counter. Consequently, in inexperienced hands the results are unreliable. While we are not prepared to state that the Owens' instrument gives errorless results, there seem to be several obvious sources of error in the Hill instrument that could be greatly reduced by increased accuracy of construction.

MARGARET INGELS (WRITTEN): There is one thought that I believe should be kept in mind when endeavoring to learn the best method for determining the amount of dust in the air. The dust determining instrument should take into account all of the dust in the air sampled. By all of the dust I mean all sizes of particles, fumes and smokes that it is possible to remove with the best of present materials and knowledge at hand. The method of testing should be as near 100 per cent as possible.

It may not be practical or advantageous to remove the very small dust particles in ventilation, but to have the determinator to remove only large particles would give distorted efficiency ratings to air cleaners.

It seems to me to be more honest to rate an air cleaner as 30 per cent efficient for removing all solid matter for a definite environment than to rate a filter as 100 per cent efficient because the determinator was no more efficient than the cleaner it was being used to test.

E. C. EVANS: I would like to ask Professor Rowley if he has noted or is going to note upward steps in the size or mesh of dirt. Manifestly there is some limit somewhere, and if we could tell, or if we knew about what was the mesh, a very definite statement could be made on efficiency. This would be some basis on which we could stand. In work of this kind in the field we note that we get very clean air, but where that air is passing into a chamber, some rotative effect in that chamber might discharge some of the dirt collected: Not, however, until there has been an accumulation in that chamber, and when tests are made on that discharged air from that chamber, the particles are rather larger than when they went in. That is, as far as you can note under a microscope. I believe if some thought could be given to this feature, we would be able to base our work on something definite.

HOMER R. LINN: I was thinking along the same line. Take Fig. 2, I wondered what effect the volume of those little chambers had on the dust count. The air quiets up after it hits the plate from the No. 1 chamber and starts into No. 2; that is, it becomes quiet, so that dust would be deposited in the chamber.

PROF. F. B. ROWLEY: Referring to Mr. Evans' discussion, it is true we will have to strike some lower limit for the size of dust particles. The magnification may be increased until there are millions of dust particles visible per cubic foot, and counting would be out of the question. From a practical viewpoint, the

method of counting dust particles must give relative values and a lower limit must be set beyond which it is not necessary to go.

Mr. Linn's criticism is well taken. It is true that some of the dust is taken from the air in its distortions around the chamber. The method was devised merely as a means of passing the air from one nozzle to another in order to find out something of the proportion of dust taken out by a nozzle. For practical purposes, I wouldn't want to advocate three chambers in series as shown.

PRESIDENT LEWIS: You are going to do some more work on that subject, aren't you, Professor Rowley?

PROFESSOR ROWLEY: Yes, and the committee would be very glad to have suggestions of any kind, either written or otherwise. The question of dust determination is a baffling one, and any suggestions as to practical, satisfactory methods would be welcomed.

No. 844

INSTRUMENTS FOR THE MEASUREMENT OF AIR VELOCITY

By J. H. PARKIN,[1] M. E., F. R. AE. S., ONTARIO, CANADA

NON-MEMBER

A DESCRIPTION is given of certain anemometers of the pressure type for measuring the velocity of air under low pressures. The construction, characteristics and limitations of the instruments are discussed and the precautions to be observed in their use are described.

INTRODUCTION

The measurement of fluid velocity is of great importance in many branches of engineering. In steam and hydraulic plants, waterworks and sewerage systems, it is necessary to know the rate of flow of water in pipes and conduits, and flowing in open channels its speed must be measured in irrigation work and stream gaging. Steam velocities are now regularly measured in power plant and industrial practice. The velocity of flow of gases, such as air, natural and illuminating gas, has commonly to be determined. In marine and aeronautical transportation the velocity of water or air relatively to the vessel or aircraft determines the speed of travel.

While the fundamental principles underlying the different methods employed for measuring the velocities of liquids and gases are generally the same, special modifications in apparatus or technique are often adopted for the measurement of any particular fluid. Thus, while certain methods of velocity measurement are peculiar to liquids and others to gases, many are applicable either to liquids or gas, as, for instance, the pitot-tube or venturi meter; while others, but little modified, may be used for either. For example, the current meter employed in river gaging differs in no way fundamentally from the cup anemometer employed for determining wind velocity.

The measuring instrument employed will depend, to a large extent, on the particular conditions, such as the nature of the fluid, its velocity and pressure, and whether the flow is in a closed pipe or open channel. Certain devices measure the general average velocity over a considerable area, while others measure

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the velocity virtually at a point. The velocity required may be that of a fluid through a pipe or channel, or that of some object through relatively stationary fluid, as in vessels and aircraft.

It is proposed to confine this paper to a consideration of certain methods for the measurement of air velocity under atmospheric or low pressure. Certain of the methods described will be applicable only to these conditions, while others, it will be apparent, may be used either directly or with minor modifications for the measurement of flow under pressure in pipes, not only of air and other gases,

FIG. 1.—PRESSURE PLATE ANEMOMETER FOR AIRCRAFT PITOT-STATIC AIRCRAFT HEAD

but also of water and liquids. The methods described will be those for the determination of velocity at a point.

DESIRABLE FEATURES IN AN AIR SPEED INDICATOR

There are certain desirable features to be sought for in selecting an instrument for use under any particular set of conditions. The instrument should be simple and direct in design, with few parts and direct connections, compact and rugged. Convenience in using the instrument is not an unimportant feature, since its absence may reduce the accuracy of the observations through fatigue. Thus, ease of reading, thereby eliminating eye strain, and direct reading, that is, indicating the velocity directly in the desired units, are desirable features.

The accuracy and sensitivity required will vary with the circumstances. The precision of the instrument, however, should be permanent, suffering no deterioration in service. Aside from the inherent accuracy of the instrument, the precision of the observations may be limited by other factors, the most important being unsteadiness or fluctuations in the velocity being measured. The effect of pulsations may be reduced by the introduction of damping in the instrument, but the greater the damping, the more sluggish it becomes, and the greater the time lag which may introduce errors in the indicated velocity.

Instruments may be self indicating or manual. Those of the former type indicate any velocity within their range without manipulation, while the latter require adjustment for each reading. If velocities of varying magnitude are to be read, the self indicating type will generally prove the more convenient; while, if the instrument is being used to enable a velocity to be held constant,

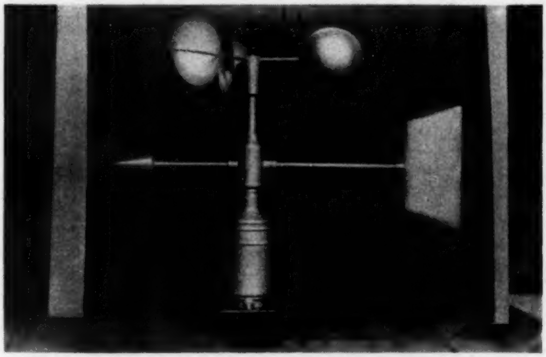

FIG. 2.—THREE CUP METEOROLOGICAL ANEMOMETER (CANADA)

there is no difference between the self indicating instrument and the one that has to be adjusted manually.

Another feature to be considered is the size of the instrument and whether it is such as to alter the conditions of flow being studied. Thus, an instrument of considerable bulk should not be employed in a small pipe where the obstruction would materially change the rate of flow.

Whether an instrument is a self standard, such that velocities can be determined directly from its reading and dimensions, or requires calibration against a standard, is perhaps not important provided that, if of the latter type, comparison with the standard is made at sufficiently frequent intervals. This, however, is seldom done. On the other hand, self standards are often less simple in construction and less convenient to use than calibrated instruments.

TYPES OF AIR SPEED INDICATOR

Air speed indicators, or anemometers, may be broadly divided into two classes, pressure type anemometers and electrical anemometers. Anemometers of the pressure type are actuated by the pressure exerted by the air due to its velocity,

while the electrical instruments depend for their action upon the dissipation of heat from electrically heated wires in the air stream. The present paper will deal only with instruments of the former type.

There are two principal kinds of pressure anemometers, namely, mechanical or moving part anemometers, such as cup or vane instruments, and pressure tube anemometers of the pitot or venturi tube type. In the former type of instrument the air reaction on a surface, vane or cup, is the active force, while in the latter the dynamic pressure of the air is employed. Most instruments of the mechanical type are practically independent of the density of the fluid, except as it affects the friction, while the indication of instruments of the pressure tube type depends on the density of the fluid.

MECHANICAL ANEMOMETERS

A—Pressure Plate Anemometers: The simplest form of anemometer employing the air reaction on a surface is that in which a plate is deflected by the

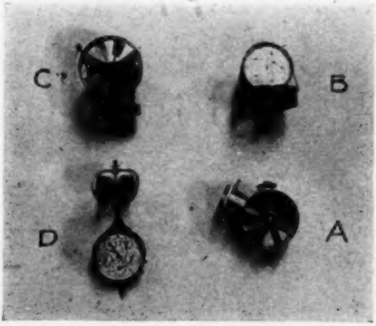

FIG. 3.—CUP AND VANE ANEMOMETERS

air flow against the resistance of gravity or a spring. Thus, in the pendulum anemometer a rectangular plate is hung on knife edges at its upper edge and loaded near the center. When placed normal to the air flow the plate is deflected until the restoring force due to gravity balances the air reaction on the plate. The angle of swing is indicated on a graduated quadrant, which may be calibrated to read in suitable units. Similar instruments using discs, or in which the plate is moved horizontally against spring resistance, have been used.

The obstruction offered by such instruments to the air flow is a serious disadvantage for many purposes, while the accuracy of the instrument is not great. In addition, the indicator is dependent on the density of the fluid. On the other hand, the instrument gives an instantaneous indication of velocity.

Recently, the principle of the pendulum anemometer has been adopted in an instrument for indicating the approximate speed of aircraft[2] as shown in Fig. 1, *A*. A small plate, about 2-in. square, is carried on the lower end of a wire arm, initially vertical, which at its upper end is coiled to form a torsion spring. The air pressure deflects the plate against the spring resistance along a quadrant

[2] Employed on the De Havilland "Moth" light aeroplane.

graduated in miles per hour. The instrument is mounted in an unobstructed position on an interplane strut where it is readily visible from the cockpits. It would seem that a similar instrument of small size and with a lighter spring might be useful for measuring roughly low air velocities in heating and ventilation and like work.

B—Cup Anemometers: Cup anemometers consist of a number, usually four, of hemispherical cups mounted on arms radiating in a horizontal plane from a vertical spindle free to rotate in anti-friction bearings, and furnished with a revolution counter. When placed in the air stream, the cup on one side is presented concave side to the air flow, while the cup on the opposite side is presented convex side to the flow. The coefficient of resistance of the cup in concave presentation being 0.665, while that of the cup in convex presentation is 0.165, the former moves with the air forcing the latter against the air flow, and the cups and spindle are thus rotated windmill fashion.

The speed of rotation, and hence the indication, is approximately proportional to the air speed, instead of the square of the air speed, as in the pressure tube instruments, and hence the cup anemometer is more suitable for measuring low air speeds.

The original so-called Robinson cup anemometer had four cups and was the standard instrument employed to measure wind speeds. Recently, as a result of a most complete study of cup anemometers made by J. Patterson of the Meteorological Service of Canada in the wind tunnel of the University of Toronto, in which a great number of combinations of cup diameters, numbers of cups, arm lengths and cup forms were investigated, it was found that a three cup arrangement has more uniform turning moment and, hence, more constant factor than the four cup. The factor is the relation between the distance moved through by the cups and that travelled by the air in a given time; in other words, it is air speed divided by cup speed. As a result, a new type anemometer having three 5-in. cups on arms 6.3 in. long has been introduced and adopted by the Canadian, American and other meteorological services. The Canadian standard anemometer is shown in Fig. 2.

Cup anemometers range in size from the large outdoor instruments used in meteorological work, such as the Kew pattern having 9-in. cups on 24-in. arms, or that described in the foregoing, to the small instrument shown at *D*, Fig. 3, having four 0.827 in. cups on 0.710 in. arms.

Cup anemometers are ordinarily of the indicating type in which revolutions of the cup spindle are indicated by needles on graduated dials, and to determine the air speed an independent time observation with a stop watch must also be made. Such instruments may have the dials directly on the instrument, the needles being driven through suitable gear trains, as in the small instrument at *D*, Fig. 3, or a contactor only may be incorporated in the anemometer with a remote electrical recording or indicating device, as in the meteorological instrument of Fig. 2. There are also tachometer cup anemometers which indicate the air speed directly.

Friction, lubrication and inertia have important effects on anemometers of this type. While every effort is made to reduce friction to a minimum by using anti-friction bearings in the large instruments and jewel pivots in the small instruments, friction still introduces inaccuracies, particularly at low speeds. In

addition, the gear trains connecting the spindle and indicating needles impose a considerable friction drag on the spindle. Curves showing percentage errors in the readings for two cup anemometers are shown in Fig. 4, and it will be seen that the accuracy of the indication is much reduced at low speeds.

Inertia of the rotating element results in too high an indication of velocity in

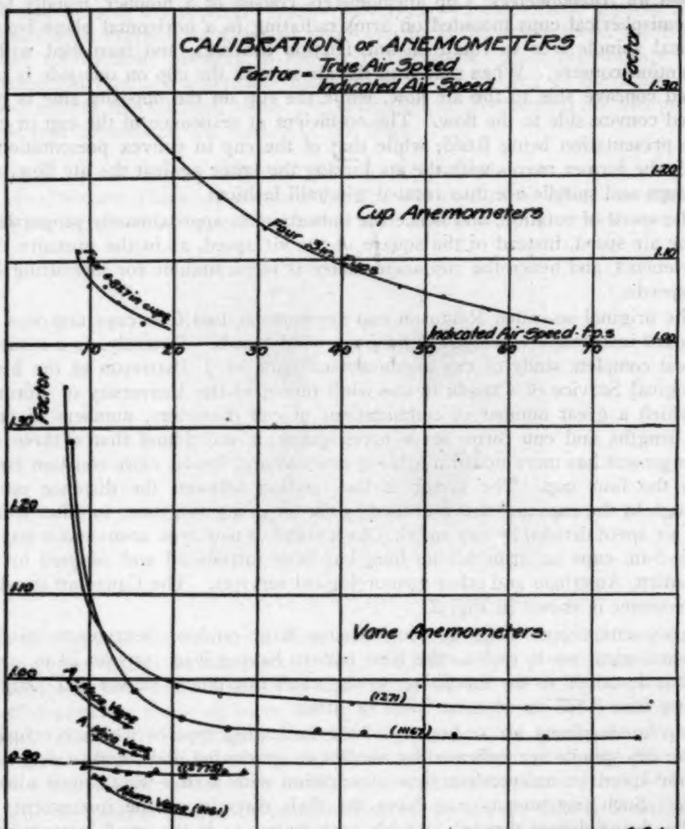

FIG. 4.—CHART

a fluctuating air current. The instrument is sluggish. Light construction minimizes this trouble.

An important characteristic of cup anemometers is that the air speed from *any* direction in the horizontal plane is indicated.

C—Vane or Screw Anemometers: This type of instrument is similar to the cup anemometer except that instead of cups, light blades or vanes set at an

angle to the plane of the arms, windmill fashion, are used. When the axis of rotation is parallel to the air flow the air reaction on the vanes has a circumferential component which rotates the spindle. Revolutions of the latter are indicated either directly on suitable dials, or an electrical contact is made after a certain number of spindle revolutions, which enables distant electrical observation to be employed.

The vane anemometer can be made of somewhat lighter construction than the cup type and, hence, is more sensitive. Mica or aluminum vanes are used and jewel bearings. This type of instrument is extensively used in ventilation and mine work. It is a delicate instrument requiring careful handling and it should not be used for velocities over 50 fps or 3,000 fpm.

The vane instrument, unlike that with cups, due to the presence of the protecting cylindrical case, requires to be more or less accurately aligned with the direction of air flow. As with the cup anemometer, a time observation is necessary to determine the velocity and friction at the spindle, and gear trains reduce the accuracy. This latter effect is shown for different vane anemometers by the curves of Fig. 4. Inertia effects are negligible with the light construction employed.

In general, one of the three positions is used for the indicating mechanism, two of which are shown at *A, B* and *C,* Fig. 3. That in which the dial case is directly behind the windmill as at *B* and *C* probably has the greatest interference effects, while that with the small case on the outside of the cylindrical casing ring, as at *A* would be expected to have least. The third position for the dials is in the center of the wheel concentric with the wheel and case and normal to the air flow. This last arrangement is more compact, attractive in appearance, and the interference probably less and more consistent than in the first position. However, the observer is more likely, with the latter instrument, to stand in such a position that he interferes with the air flow through the instrument.

PRESSURE TUBE ANEMOMETERS

Anemometers of this class may be divided into pitot type and venturi type instruments.

A—Pitot-Tube: The pitot-tube is simply an open ended tube facing the current. The air, or other fluid, flowing into the open end is brought to rest and thereby exerts a pressure on the air in the tube equal to the dynamic pressure. The latter is expressed by the relation $\rho\frac{v^2}{2}$, ρ being the mass density of the air or fluid, *i.e.,* $\frac{w}{g}$ where w is the weight in pounds per cubic foot and v in feet per second. The pitot-tube is commonly employed in conjunction with a second tube in which the pressure is equal to the static pressure in the air stream, so that the pressure difference between the tubes is the dynamic or velocity pressure. The combination is known as a pitot-static tube, or, less correctly, simply as a pitot-tube.

It has been found that the form of the open end of the pitot-tube is not important since, with forms as widely different as a hole in a disc and a finely tapered tube end, the pressure in the tube is found to be the correct dynamic

pressure. On the other hand, great care must be taken in forming the end of the static tube if the true static pressure is to be obtained in the tube. The most satisfactory arrangement consists of one or more holes (static openings) of diameter not greater than 0.04 in. drilled in the walls of a tube whose axis is parallel to the air stream. The end of the tube is closed and faired or pointed so that the air flows in smooth parallel streams past the static openings. It is important that the latter should be of small diameter, drilled normal to the surface and with all burrs removed.

In some instruments the pitot and static tubes are quite separate, as in the aircraft air speed head, shown at *B*, Fig. 1. Commonly, however, the pitot and

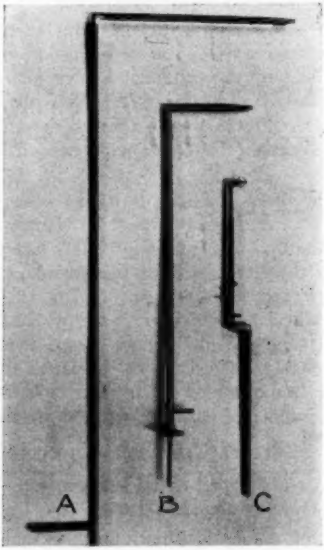

FIG. 5.—PRESSURE TUBE ANEMOMETERS

static tubes are combined in one instrument as shown at *A* and *B*, Fig. 5. These instruments each consist of two concentric tubes of about 3/16 in. and 5/16 in. O. D. The smaller, inner tube is the pitot-tube extending from the pitot, or impact opening, to the corresponding pressure connection. Surrounding this tube is the larger static tube in which, near the nose, the static openings are drilled. The static openings should be 2 or 3 in. (not less than six diameters) back of the tapered nose and about 10 in. ahead of the heel or shank. The nose should be finely tapered, not to improve the dynamic opening, but so that the air will flow smoothly past the static openings at all speeds, thus permitting the true static pressure to be obtained.

The theory of the pitot-tube is well known. It is commonly developed on a basis of Bernouilli's theorem, considering the air incompressible, but more correctly from the principles of thermodynamics, taking into account the com-

pressibility of the air. In the latter case for adiabatic compression, the pressure on the impact opening is

$$p=p_0\left(1+\frac{(\gamma-1)\,\rho_0 v^2}{2\gamma\, p_0}\right)^{\frac{\gamma}{\gamma-1}}$$

so that a pitot-static tube transmits to a differential gage the difference between this pressure and the normal static pressure, or

$$p-p_0=\frac{\rho_0 v^2}{2}+\frac{1}{\gamma-1}\,\frac{\rho_0 v^2}{2}+++$$

In this expression all terms after the first are negligible to a precision of 1 per cent for speeds below 225 fps and the equation reduces to that developed on a basis of Bernouilli's theorem.

The pitot-static tube, constructed as described in the foregoing, therefore, requires no calibration, the equation $p=\frac{\rho v^2}{2}$ applying with an accuracy of 1/10 per cent of the velocity up to 50 fps (3,000 fpm) and 1 per cent up to 225 fps (13,500 fpm). At a speed of 40 fps (2,400 fpm) the pressure difference between the tubes is only 0.365 in. of water, and this small pressure together with the fact that it varies as the square of the speed, and hence is very small indeed at low speeds, are the principal disadvantages of the instrument.

Owing to the small magnitude of the pressure difference for ordinary air velocities, various modifications have been introduced to increase the reading. In one case an aspirator tube replaces the static tube. The aspirator tube may simply be a pitot-tube with open end downstream, as in the so-called double pitot, or the open end of a tube pointing upstream is covered by a small conical cap with open base downstream. The pressure inside the cone is then less than the normal static pressure. This construction is employed in the Clift aircraft head and in the yawmeter shown at *C*, Fig. 5.

No theory has been developed for this type of tube, as it involves a consideration of the external flow about the tube and the negative pressure at the rear. The pressure difference between the two tubes may be 40 per cent greater than that between the tubes of the ordinary pitot-static tube, and if the formula of the latter is used, a coefficient must be used, ranging in value from 1.35 to 1.40.

The pitot-static tube is perhaps one of the most commonly used instruments for the measurement of the velocity of fluids which are clean and not too viscous. It is simple, inexpensive, compact, convenient, of high accuracy and, if properly constructed, requires no calibration. Owing to the small magnitude of the velocity head, the instrument is not satisfactory for measuring velocities below 20 fps (1,200 fpm) for which the velocity head is less than 1/10 in. of water.

There are certain precautions to be observed in the use of the pitot-static tube if accurate results are to be obtained. It is important that the tube be set parallel to the air flow. A misalignment of 4 deg will cause an error of 1 per cent and 17 deg misalignment about a 5 per cent error in the reading. The air stream immediately behind the instrument must be free from obstruction. The effect of an obstruction here is to cause the lines of flow to be deflected, and if this

deflection occurs at, or upstream, from the static openings, the true static pressure will not be obtained. For this reason the static openings should be well forward, some 8-10 in., of the shank of the instrument, and if the tube is carried on a rod or other support, the latter should be kept well back of the instrument. A hemispherical clamp, 1 in. in diameter, 3 3/16 in. behind the static openings of the tube, shown at *B*, Fig. 5, was sufficient to cause a 1 per cent error in the velocity indicated. For the same reason the tube should not be used too close to a wall or other object, since confining the air flow between the wall and tube will result in an error in the static pressure obtained.

B—Pitot-Venturi Tube: The static tube in the pitot-static instrument is, in some cases, replaced by a venturi tube in order to increase the magnitude of the pressure difference obtained. The double conical form of the venturi tube is familiar to all engineers, and it is well known that a fluid flowing through the meter experiences as a result of the reduction in cross-sectional area in the

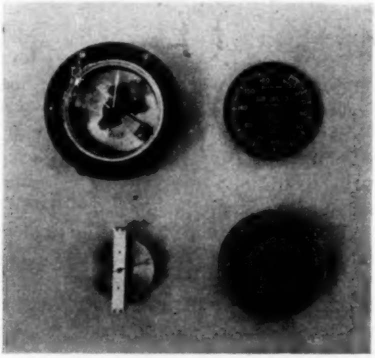

FIG. 6.—MECHANICAL GAGES

tapering cone, an increase in velocity, accompanied by a corresponding decrease in pressure. Thus, the pressure at the throat is less than the normal static pressure, and hence the pressure difference between the dynamic pressure of the Pitot tube and the reduced pressure at the throat of the venturi tube will be much greater than for the usual pitot-static combination.

The venturi tube may be employed alone, using the pressure difference between the throat and the normal static pressure, but generally, for the foregoing reason, it is used with the pitot tube.

While the theory of the venturi tube inserted in a pipe line is well known, no theory has been developed for the venturi tube in free air when the flow around the outside must also be considered. Considerable research has been done on venturi tubes used in this way, however, and the following points have been determined:

1. The entry cone should be of about 20 deg angle and from ¾-1 in. long.
2. An exit cone angle of 4 deg 50 min and a length of 6 in. are satisfactory.
3. Flaring the base of the exit cone increases considerably the suction.
4. The throat between entry and exit cones should be short, not exceeding 0.03 in.

5. The efficiency of the tube drops with increase in size, a throat diameter of from $\frac{3}{8}$ to $\frac{1}{2}$ in. being most satisfactory.
6. There is no advantage in the use of cones with variable angle, *i.e.,* curved surfaces.

The suction obtained can be much increased by using a double venturi tube. A small venturi is arranged in the entry cone of the larger tube, with the end of its exit cone at the throat of the larger tube. The reduced pressure set up by the larger tube serves to increase the flow through the smaller, thereby developing in the throat of the latter a much greater suction. Suctions eight times that of the single tube have been obtained in this way.

Pitot-venturi tubes, both single and double, are used to a considerable extent for indicating the speed of aircraft. The American and French Air Services employ this type of instrument. It would seem to have possibilities as an instrument for use in heating and ventilation work because of the large pressure difference, although its form renders it less convenient to use than the pitot-static tube.

The same precautions must be observed in using this instrument as for the pitot-static tube. The venturi must be kept at least 25 diameters ahead of any obstruction, even if of fairly stream-lined form, such as an aircraft strut, to prevent blanketing of the tube and loss of suction.

MANOMETERS

Anemometers of the pressure tube type require some form of pressure gage to measure the pressure difference between the tubes, and because of the small magnitude of this pressure difference, such gages must possess great sensitivity. A sensitivity of 0.0005 in. of water is desirable, although 0.001 in. is often sufficient. The range required will generally not exceed 3 in. of water, or at the outside 6 in., since the air velocity corresponding to a dynamic pressure of 3 in. of water is 115 fps (6,900 fpm).

The gages should be free from indeterminate errors due to capillarity, viscosity, hysteresis, friction and inertia. Adjustable damping is desirable to enable the velocity in unsteady flow to be measured. Simplicity and convenience are of much importance. The use of colored liquids, reading of one meniscus only, zero adjustment and scales reading directly in the desired units, all facilitate the taking of the readings and improve their accuracy through eliminating fatigue and needless computations. It is also desirable, as previously noted, that the gage should be a self standard, if this can be effected without too great sacrifice in convenience or other features.

The gages employed are usually either mechanical or gravity.

A—Mechanical Gages: In the mechanical type the movement of a diaphragm, or other similar device, under the action of the pressure difference, is mechanically magnified and communicated to a needle moving on a dial. While non-metallic diaphragms of such material as rubber or parchment have been used, a thin corrugated metal diaphragm or bellows is now most commonly employed, the arrangement being very similar to that of the ordinary aneroid barometer. Gages of this type, in conjunction with pitot-static or pitot-venturi tubes, are standard equipment on aircraft for indicating air speeds. Two such gages are shown in Fig. 6, one having a corrugated metallic diaphragm, the other an oiled silk diaphragm.

Mechanical gages are less simple in principle and more complicated in construction than those of the gravity type. They are subject to errors due to friction in the pivots and hysteresis in the diaphragm. Obviously, such gages require calibration.

B—Gravity Gages: The gravity type gages employed for velocity measurement purposes are basically simply *U* tube manometers, and hence are usually termed manometers or micromanometers. Various modifications are introduced in the simple *U* tube to magnify the reading and so render the gage more sensitive. These gages are made either self indicating or manual in operation.

(1) SELF INDICATING MANOMETERS.

In manometers of the self indicating type two methods of magnification are employed.

a—Two Liquid Differential Gage: In the two liquid differential gage, two non-miscible liquids of nearly the same specific gravity are employed in such a

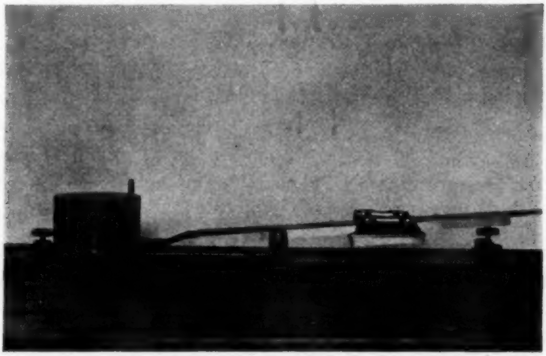

FIG. 7. SLANTING (KRELL) MANOMETER

way that the difference of their specific gravities is the operative factor. Additional magnification is obtained by enlarging the upper portions of each leg of the *U* tube into cisterns of about 100 times the cross sectional area.

Alcohol, colored with aniline dye and kerosene, are the two liquids generally used, although water or methylated spirits may be used with the kerosene. The specific gravity of the alcohol may be adjusted by the addition of water to be as nearly that of the kerosene as desired. If the specific gravities are too nearly equal (less than 0.05 difference) the gage is difficult to manipulate, as the meniscuses become irregular. Methyl alcohol and gasoline and amyl ether and water have also been used. Care must be taken that there are no constituents in the one soluble in the other, and this may be guarded against by keeping the two in the same container before using.

The formula for the gage in which the lighter liquid is in one leg only of the *U* tube is

$$P_1 - P_2 = R\ (s_2 - s_1) + \frac{A_t}{A_r} R\ (s_2 - s_1)$$

and if in both legs of the *U* tube the expression is

$$P_1 - P_2 = R\ (s_2 - s_1) + \frac{A_t}{A_c} R\ s_1$$

where R is the movement of the surface of separation. A_t and A_c areas of tube and cistern, respectively. s_2 and s_1 specific gravities of heavier and lighter liquids, respectively.

With alcohol and oil of specific gravities 0.834 and 0.79, respectively, a magnification of 17-18 is secured with oil in one leg and one meniscus only has to be read, while with oil in both legs the magnification becomes about 23, but two meniscuses have to be observed.

The tubes should be of ⅛ to ¼ in. bore and length 30 times the desired range in water. With large bores the meniscus becomes indeterminate, and with small bores it may break. The longer the tubes the greater the inertia and consequent lag. With ⅛ in. by 30 in. tubes the lag may be 1 min.

In tests made at the University of Toronto this gage was found unsatisfactory. It suffers from errors due to capillarity, viscosity, inertia, hysteresis and imperfections in the tubing. Its precision is low. The accurate determination of the specific gravities of the liquids is an additional disadvantage.

b—Slanting Manometer: In the slanting or Krell manometer, magnification is obtained by inclining one leg of the *U* tube at a small angle to the horizontal, so that for a given vertical displacement of the meniscus, the movement along the tube is much increased. Angles of 3 deg and 6 deg to the horizontal are ordinarily used, giving magnifications of 20 and 10, respectively.

The gage is rendered more convenient by replacing the vertical leg of the *U* tube by a reservoir or cistern having several hundred times the cross sectional area of the slanting tube. This makes it necessary to observe only one meniscus, that in the slanting tube, since that in the reservoir remains substantially constant.

The slanting tube should be uniform in bore, about ¼ in. diameter, and straight.

Alcohol (colored) has been found best for this gage. The precision when employed with a pitot-static and the velocity calculated from the dimensions is about 1½ per cent, but as the gage responds to velocity changes of ¼ per cent, if calibrated against a standard, it may have the precision of the standard.

A gage used with success at Toronto is shown in Fig. 7. The glass tube is clamped with transparent celluloid bands into a *V* groove machined in a flat steel bed. The latter is pivoted at the lower end and provided with an elevating screw on a trunnion at the other end. Two steel wedges, machined to angles of 3 and 6 deg, respectively, attached to the bed, carry spirit levels. By adjusting the elevating screw until one or other of the spirit levels is level, the inclined tube is brought to a slant of 3 or 6 deg to the horizontal, irrespective of the position of the base of the gage.

The *V* groove keeps the tube straight, and by marking the glass tube it can always be returned, after cleaning, to the position it occupied when calibrated. A strip of white paper behind the glass tube improves the visibility greatly, and by bevelling one edge of the graduated scale to a knife edge and

fitting it to project halfway over the glass tube, errors in observation, due to parallax, are much reduced.

The reservoir is of spun copper 5 in. in diameter by 3 in. deep, and should be insulated to eliminate temperature effects. The reservoir is provided with an elevating screw so that the meniscus may be adjusted to zero on the scale.

This particular gage has been found most satisfactory for many purposes. It is simple, inexpensive, precise, consistent, very quick to respond and convenient to read. Errors due to capillarity and inertia are small.

(2) MANUAL ADJUSTMENT MANOMETERS.

In manometers of the manual type, instead of permitting the liquid to flow from one leg of the *U* tube to the other to balance the applied pressure difference, one leg of the *U* tube is raised bodily, without allowing the liquid to flow in the tube, to establish the difference in level of the liquid necessary to balance the pressure difference. The distance through which the leg is raised

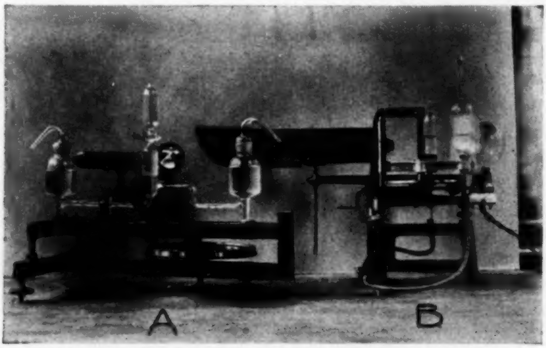

FIG. 8. CHATTOCK MICROMANOMETERS

from the zero position to maintain equilibrium is then equal to the applied pressure difference. Troubles due to capillarity are thus avoided, since theoretically there is no movement of liquid in the tube. To insure that there is no such movement there is incorporated in these manometers some form of sensitive indicating device designed to show the slightest movement of the liquid. The more sensitive this device the greater the sensitivity of the manometer.

a—Chattock Micromanometer: One of the best known gages of this type is the Chattock tilting gage shown at *A,* Fig. 8, which is largely used with the pitot-static tube in aerodynamic laboratories. The gage consists of a *U* tube in which each leg is enlarged to form a bulb of about 2 in. diameter. The flow indicating device is located in a vertical tube about 1 in. in diameter, midway between the bulbs. The tube from one leg or bulb is carried up centrally in the 1 in. tube, terminating in an open end, the edges of which are ground off to a knife edge. The tube from the other leg enters the side of the 1-in. tube. The upper part of the 1-in. tube is filled, from a reservoir above it, with oil to a level below the top of the central tube. The remainder

of the gage is filled with water. The water rises in the central tube and forms a meniscus or bubble in the oil at the top. Evidently, the slightest flow of water from one leg of the *U* tube to the other will distend or contract this bubble. A microscope fitted with a hair line is focused on the bubble with the hair line tangent to the latter, and enables minute movements to be detected.

The whole of the glassware described in the foregoing is carried on a tilting frame controlled by a micrometer screw and dial.

The applied pressure difference tends to displace the water from one leg of the *U* tube to the other, this distends the bubble and the *U* tube is tilted by means of the micrometer screw until the head, due to the difference in level of the two bulbs, balances the applied pressure, as is shown by the bubble remaining tangent to the hair line of the microscope. From the dimensions of the gage and pitch of the screw, the change in elevation, and hence the pressure, may be calculated.

The center to center distance of pivots and bulbs should be known to within 0.001 in. and the pitch of the screw to within 0.0001 in.

In the original Chattock gage castor oil was employed, and to prevent it clouding through contact with the water, brine was used instead of water. This proved awkward, necessitating the determination of the specific gravity of the brine. In the gages now used at Toronto, distilled water and a medicinal paraffin oil have been used with success.

The Chattock gage is very sensitive. A pressure change of 0.00025 in. of water can be readily detected. It is an absolute standard. While temperature changes affect the accuracy, the effect for ordinary temperature changes is small. The chief disadvantages of the gage are the fragile nature of the glassware, the extreme care necessary in use to avoid rupturing the bubble, and the eyestrain involved in long continued observation through the microscope.

b—Modified Chattock Gages: The first and last of the difficulties mentioned in the previous paragraph have been eliminated in a modified design developed in England. Gages of the modified design, as made and used at Toronto, are shown at *B*, Fig. 8. Instead of the tilting arrangement, one bulb is raised vertically on a micrometer screw. The bubble indicator and the second bulb are combined to form the other leg of the *U* tube. The two legs are connected by a rubber tube in which a glass stop cock is inserted. The glassware is thus simple.

Instead of the microscope, the enlarged images of the bubble and hair line are projected on a ground glass screen rendering observation easy. The bubble is illuminated by means of a concentrated filament stereopticon lamp. Spirit levels are fitted to enable the gage to be properly levelled.

It is evident that the accuracy of this gage depends solely on the accuracy of the micrometer screw. The gage is as accurate as the screw. The gages used at Toronto are sensitive to 0.0001 in. of water. While this type of gage, particularly in the latter form, is an excellent laboratory instrument, it is probably too delicate and sensitive for use in ordinary engineering practice.

c—Direct Lift Gage: A more satisfactory form of manometer for general use, possessing almost equal accuracy, and of simpler and more robust construction, is the so-called direct lift gage, shown in Fig. 9.

In this gage a large spun brass reservoir forms one leg of the *U* tube, and is connected through rubber tubing to a short length of glass tubing mounted on a metal bracket, which can be rotated about a horizontal pivot, so that the tube may be set at any inclination to the horizontal (as in the slanting Krell gage). The pivot is carried by a block which can be raised or lowered by means of a micrometer screw and dial. The gage is filled with colored alcohol whose specific gravity must be accurately determined.

The meniscus in the slanting tube, with the gage at zero, is adjusted by raising or lowering the brass reservoir (using screw below) until it is tangent

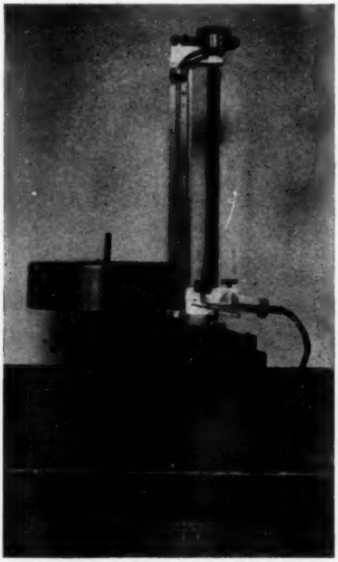

FIG. 9.—DIRECT LIFT GAGE

to a hair line engraved on glass mounted in front of the tube. Parallax is avoided by using a mirror behind the tube.

The alcohol, under the action of the applied pressure difference, tends to rise in the slanting tube, and this tendency is counteracted by raising the tube bodily by means of the micrometer screw, keeping the meniscus tangent to the hairline. The amount the tube is elevated is then the pressure difference in inches of alcohol. The sensitivity of the gage may be varied by adjusting the slope of the inclined tube.

This manometer is evidently a self standard whose precision depends only on the accuracy of the micrometer screw. The use of the large reservoir renders it necessary to observe one meniscus only. The range of the gage can be made quite large, that illustrated having a range of 10 in. of alcohol.

The direct lift gage should prove very useful in heating and ventilation work for the accurate measurement of fairly constant pressures and velocities.

D—Wahlen (Illinois) Gage: The Wahlen gage developed at the University of Illinois, is similar in form to the direct lift gage, but employs a different indicating device. The tube connecting the two bulb arms of the *U* tube is bent upward to form an inverted *U* tube of which one arm is enlarged and the other constricted to a very fine bore (3 mm). The upper part of the inverted *U* is filled with a mixture of kerosene and ligroin, the surfaces of separation between the mixture and the alcohol, with which the rest of the gage is filled, being located in the enlarged and constricted sections of the inverted *U*. The slightest movement of the liquids in the instrument under the action of a pressure difference is then greatly magnified in the movement of the meniscus in the fine bore tube from a reference line engraved on the latter. This movement is counteracted by raising one bulb of the main *U* tube by means of a micrometer. A movement of the micrometer screw of 0.001 in. moves the meniscus 1/16 in. in the fine bore tube, if the difference in specific gravity of the kerosene-ligroin mixture and alcohol is adjusted to be 0.0085. The necessity of maintaining the difference in specific gravity of the two liquids within rather narrow limits is emphasized in the description of the gage and would appear to be a disadvantage. It is stated that the gage is accurate and sensitive to a pressure difference of less than 0.0001 in. of water.

DISCUSSION

Dr. E. Vernon Hill: The paper is a fine description of instruments and methods. If we discussed the pitot-tube or anemometer, we might spend the whole morning at that, especially if we got into this discussion of how to use the anemometer, which has been stirring up Chicago for the past four or five months.

President Lewis: This is an important subject, because I do not think we have as accurate instruments for measuring air velocities and quantities as we should have, and I hoped that through this discussion there might be some light thrown on the subject of development.

W. A. Rowe: The success of the use of the anemometer, besides the proper technique in its use, undoubtedly depends on the accuracy of calibration. From observations in the use of anemometers, more in coal mines than in buildings, where they are in almost daily use, it seems to me the greatest need is a convenient means for easy calibration, and I have suggested to one instrument man that if they could develop such a convenient, easy method it would be a great boon to those who have to use that instrument. Take a group of mines, where the anemometer is used so continuously, you will find that it is quite difficult to have to send it to some remote place and have it calibrated and returned back without a chance of damage to the instrument and without delay.

I know of some very interesting developments which are coming along right now, which will be brought out publicly in the near future, and that is the substitution of resistance wire for the anemometer for measuring extremely low air velocities, such as are encountered on register faces.

Prof. F. E. Giesecke: We have recently completed an investigation to determine an acceptable method of ventilating hoods in chemistry laboratories.

The problem was to determine the arrangement of baffle plates at the back of the hood so as to secure a uniform distribution of air over the face of the hood. We solved our problem by stringing horizontal wires, about 12 in. on centers, across the face of the hood, and suspending, from the wires, paper flags about 4 in. by 6 in. The inclinations of the flags indicated the velocity of the air at the respective locations; the baffle plates were adjusted until all flags had the same inclination. The volume of air passing through the hood was measured by means of a device built according to a design by the National Air Filter Co.

PROF. A. P. KRATZ: We have had some experience at the University of Illinois with measuring air, particularly with the anemometer, and we find that the calibration in the anemometer must be made under exactly the same conditions as those for which it is going to be used. Any calibration made under artificial conditions will hardly apply under the conditions of use. For some of our work we have developed a tank on scales in which we weigh the air and calibrate the anemometers by making traverses of the register faces under exactly the conditions of use. In other cases we calibrate the anemometers against a Pitot tube, reducing the area of the pipe in which the Pitot tube is placed until we get velocities high enough to be measured by the Pitot tube, and in every case using at least twenty-point traverses with a Pitot tube.

The point I wish to leave with you is that there is no easy way of calibrating air-measuring instruments. They have to be calibrated under the conditions of use.

F. C. HOUGHTEN: The paper is a very valuable one in that it brings together various possible methods of measuring air flow. It is particularly timely for the entire heating and ventilating industry is very much interested in the subject at this time. In measuring air flow we can generally divide the work that may be done into two classes: measuring air flow in a confined space or duct and measuring air in the open, such as the discharge through a grille.

The subject recently came to the attention of the laboratory through a study of the measurement of air flow through registers and grilles made by the *Chicago Sheet Metal Contractors' Association*. A cooperative arrangement is being perfected now whereby the Research Laboratory, Armour Institute, and the *Chicago Sheet Metal Contractors' Association* will make a study of this entire subject. From the preliminary work already carried on there is an indication that we will get some very valuable data. The preliminary work already carried on shows that the method which the Society has been advocating is from 30 to 40 per cent in error, and that is quite a serious error when you consider that it is in the measurement of the air delivery of a ventilating system in a building. That error in method has continued for a number of years, and is another example of the fact that very frequently when an individual or authoritative organization takes a stand on a subject, the general public forgets about it and takes it for granted that because such a statement has been made by such an authority, it is a fact. Such cases, as a rule, hinder scientific progress.

W. H. CARRIER: I want to emphasize just what Mr. Houghten has said, not merely in reference to anemometer testing alone, but the whole thing of

having a committee, a technical committee to decide on what ought to be done, and then accepting the thing as a code, when we have a Laboratory. That was done before we had a Laboratory, but now that we have a Laboratory, I think we should be much more cautious than we have been in the past, and this is a very good indication of why we should be cautious in accepting data because, academically, we think it ought to be so, and so, rather than basing it on real tests at our Laboratory. That is where the Bureau can render valuable service. We have not gone into this question on anemometer air measurements before. That is a much needed thing.

President Lewis: We appreciate that suggestion and will refer it to the Research Committee.

E. D. Milener: Emphasizing what Mr. Carrier just said, when testing gas furnaces in the *American Gas Association,* we ran up against the question of the accuracy of measurements of air passing through furnaces and through the ducts. At first they were based on anemometer readings, Pitot tube readings, etc., but we realized that we were 20 or 30 per cent inaccurate, which checks with the experience of one of the previous speakers. We then decided that the only way to get accurate measurements was by the Thomas meter, so the *American Gas Association* Laboratory has now very well equipped apparatus which depends upon the volume of measurements of all the air passing through the furnaces and through the ducts being measured by means of the Thomas meter, which is accurate to 1 per cent.

No. 845

FIVE SUGGESTED METHODS OF APPRAISING INSULATIONS

By Paul D. Close[1], New York, N. Y.

MEMBER

THE insulation industry is growing rapidly. New materials are being developed constantly, and the insulation idea is becoming more and more popular through intensive advertising and promotion. Unfortunately, however, the public in general does not have sufficient knowledge of the meaning of the word insulation to be able to differentiate between insulating materials and those not entitled to this classification. The result is that many unwarranted claims are being made by concerns endeavoring to capitalize on the promotional work being done at the expense of the ignorance of the public of the subject of insulation.

As an example, exceptional insulating qualities are claimed for a new building paper, but it is a fact well known to heating engineers that the heat resistance of building paper is so small that it may safely be neglected in heat transmission computations. Building paper is, of course, intended primarily to reduce air infiltration through a wall.

An imitation marble about ⅛ in. thick is claimed to have high resistance to the passage of heat and *cold.* It is contended that a certain plaster board used in place of ordinary materials will result in a substantial saving in fuel and reduction in the amount of radiation required for a building. It is also stated that this product is more impervious to infiltration than other materials, although recent tests indicate that infiltration through almost any plastered wall is a negligible quantity.

A corrugated paper ⅛ in. thick is promoted for insulation purposes, and it is claimed that thickness is not essential with this product. There are also other materials for which extravagant claims are made, which, although having relatively low conductivities, are not installed in sufficient thicknesses to result in an appreciable reduction in the heat transmission.

In addition to the many unwarranted claims being made, there is an endless amount of controversy over small differences in conductivities of insulations. Such differences are usually within the error of testing, and are insignificant when compared on the basis of over-all transmissions. It is certain

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[1] Technical Secretary, American Society of Heating and Ventilating Engineers.

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that they are of far less consequence than other variables entering into heating calculations. This chaotic state of affairs suggests the need for a means of grouping or classifying insulations and for rating them.

The following suggested methods are designed to serve this purpose, but should not be construed as standards, as they are merely offered for preliminary consideration to encourage further thought on the subject.

TABLE 1—SUGGESTED CLASSIFICATION OF INSULATIONS ACCORDING TO CONDUCTIVITIES

Rating	Conductivity Range (Btu per hr per sq ft per 1 F per 1 in.)	Types of Insulation		
		Rigid	Flexible	Fills
Class A	0.30 or under	7 lb Corkboard 10.6 lb Corkboard Torfoleum	Balsam Wool Cabots Quilt Dry Zero Hairinsul Hair Felt Keystone Hair Linofelt Themofelt (Hair and asbestos)	Rock Wool Sprayo-flake
Class B	0.301 to 0.35	7.3 lb Balsa Wood Celotex 14 lb Corkboard Inso Board Insulite Masonite Maftex	Fibrofelt Flaxlinum	Cranulated Cork 18 lb Thermofill
Class C	0.351 to 0.40	8.8 lb Balsa Wood Homasote Lith Rock Cork Thermosote	Themofelt (Jute and asbestos)	
Class D	0.401 to 0.50			12 lb Insulex 12 lb Pyrocell 24 lb Thermofill
Class E	0.501 to 0.60	20 lb Balsa Wood Magnesia		18 lb Insulex 18 lb Pyrocell 26 lb Thermofill 34 lb Thermofill
Class F	Above 0.60	Not Classed as Insulations.		

GROUPING OF INSULATIONS BY CONDUCTIVITIES

The first suggested method of classifying insulations is to group them according to their conductivities as shown in Table 1. By this method, all insulations having conductivities of 0.30 Btu per hour per square foot per degree Fahrenheit per inch of thickness or below, would be designated as Class *A* materials; those between 0.301 and 0.35, Class *B* materials; 0.351 to 0.40, Class *C;* 0.401 to 0.50, Class *D;* and 0.501 to 0.60, Class *E*. Materials having conductivities

higher than 0.60 would not be classified as insulations. The classifications given in Table 1 are in most cases based on conductivities published in Table 5, Chapter 1, of the A. S. H. & V. E. GUIDE, 1929, using U. S. Bureau of Standards values where tests by this authority have been made.

It will be noted that the materials in Table 1 are grouped according to types of insulation, namely, rigid insulations, flexible insulations and fills. Cork board and the board forms of insulation are classified as rigid materials; the so-called felts, quilts and soft materials are designated as flexible insulations, and the powdered, flaked and aerated materials usually confined between the studding, joists, rafters or furring strips are designated as fills. Thus, pure cork board having a density of 7.0 lb per cu ft would be a Class *A* rigid insulation, and hair felt a Class *A* flexible insulation. A cellular gypsum fill having a density of 18 lb per cu ft, and a conductivity of 0.59 per in. of thickness, would be a Class *D* fill.

One of the fallacies of the foregoing system is that it does not afford a true comparison between materials of different types of the same or different conductivities installed in different thicknesses. Neither is the manner of installation taken into consideration by this method. A Class *C* fill, if installed between the studding in a thickness of 3⅝ in. may reduce the heat transmission of a certain wall to a greater extent than a Class *B* rigid insulation, or a Class *A* flexible insulation. Similarly, a Class *B* ***flexible*** may be more effective than a Class *A* ***rigid***, due to the manner of installation.

COMPARISON BY RESISTANCE METHOD

The second proposed method which has no bearing on the first one, would classify insulations according to the total heat resistance they add to a given type of construction, thus giving consideration to the conductivity as well as the thickness installed. Suggested classifications according to this method are given in Table 2. A 1-in. thickness of a rigid insulation having a conductivity of 0.33 Btu per hour per sq ft per deg Fahrenheit per inch of thickness, would add a resistance of $\frac{1.0}{0.33}$ or 3.03 and would therefore be a Class *H* insulation. A 1-in. thickness of a flexible insulation having a conductivity of 0.27 Btu per hour per square foot per degree Fahrenheit per inch of thickness, would add a resistance of $\frac{1.0}{0.27}$ or 3.70, and would therefore be given the same classification in this case. If a fill having a conductivity of 0.50 Btu per hour, per square foot, per degree Fahrenheit per inch of thickness, is installed between 2 x 4 in. studding, the actual heat resistance of the thickness installed would be $\frac{3.625^{2}}{0.50}$ or 7.30, and this material would then be given the Class D rating according to Table 2. The chart (Fig. 1), which is based on the ratings given in Table 2, may be used for determining the resistance and classification of any insulation according to the conductivity and thickness installed.

[2] 3.625 represents the actual width in inches of 2 x 4 in. studding.

Fills are usually installed so that an air space is entirely filled, thereby cancelling the insulating effect of the air space, which, for purposes of comparison

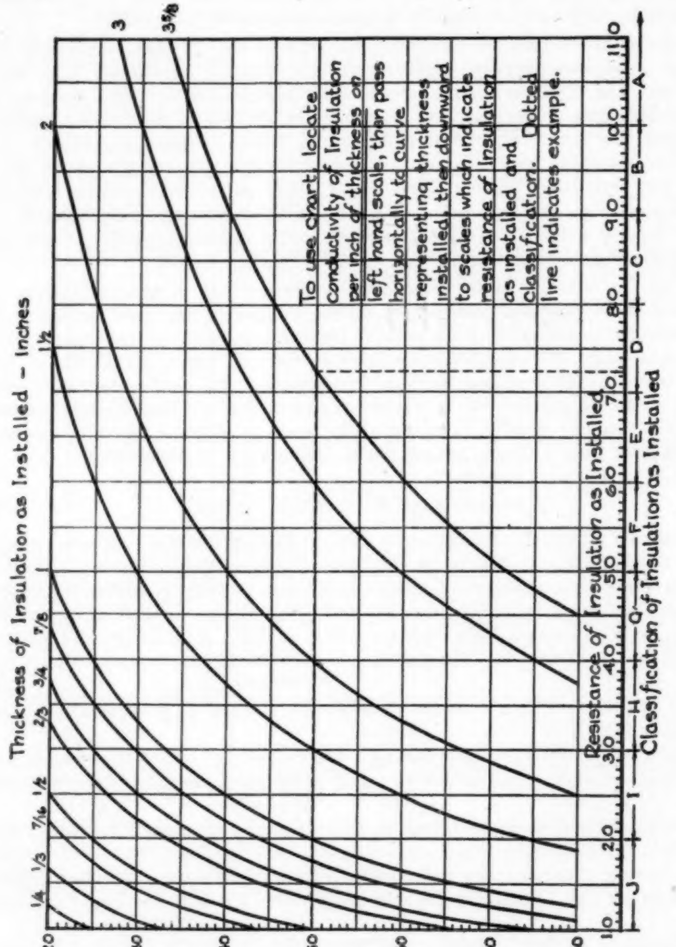


FIG. 1. CHART FOR DETERMINING CLASSIFICATION OF INSULATIONS ON BASIS OF HEAT RESISTANCE INSTALLED.

should be deducted from the resistance of the installed thickness of the insulation. On the other hand, rigid and flexible insulations may be applied so as to create one or more additional air spaces, or a rigid insulation may be used

in place of other building materials. To be strictly correct, therefore, any comparison on the basis of resistances should also take these facts into consideration.

PERCENTAGE REDUCTION IN HEAT TRANSMISSION

The resistance method of comparison or classification may be objected to on the grounds that such a method is an indirect one and does not indicate the true relationship between two materials as installed from the standpoint of effectiveness.

Resistances afford an easy method of comparison because they are additive, but unfortunately they do not tell the complete story. Neither are conductances (the reciprocals of resistances) of the actual thicknesses installed (Table 3 and 4) correct indices of the relative merits of insulations, even considering air spaces and other factors. Hence, it seems logical that exact comparisons should be made between over-all transmissions of insulated and uninsulated constructions.

TABLE 2—CLASSIFICATION OF INSULATIONS ACCORDING TO HEAT RESISTANCE OF THICKNESS INSTALLED

Rating	Heat Resistance Added
Class A	10.01 or more
Class B	9.01 to 10.00
Class C	8.01 to 9.00
Class D	7.01 to 8.00
Class E	6.01 to 7.00
Class F	5.01 to 6.00
Class G	4.01 to 5.00
Class H	3.01 to 4.00
Class I	2.01 to 3.00
Class J	1.01 to 2.00

The third suggested method of classification, therefore, involves the principle that the degree of insulation obtained approaches 100 per cent as the heat transmission coefficient approaches zero. In other words, a material installed in a sufficient thickness or having a sufficiently low conductance for the thickness installed to reduce the heat transmission to a value approaching zero, would have an insulating value approaching 100 per cent. This relationship is expressed by the formula:

$$E = 100 \left(\frac{U_o - U_i}{U_o}\right) \text{ per cent}$$

where—

E = Efficiency of the thickness of insulation installed, or the percentage reduction in heat transmission.[3]

U_o = Coefficient of transmission of the uninsulated construction;

U_i = Coefficient of transmission of the insulated construction.

[3] The value of E is not an index of the percentage saved of the fuel or radiation required to heat a building. The percentage saved of the total heat losses obtained by insulating the walls and/or roof of a building must necessarily involve all of the heat losses, including the infiltration losses and the transmission losses through glass.

TABLE 3—CONDUCTANCES OF VARIOUS THICKNESSES OF INSULATIONS BASED ON TESTS CONDUCTED AT U. S. BUREAU OF STANDARDS AND COMPUTED FROM CONDUCTIVITIES PER INCH OF THICKNESS

Material	Description	Density (lb per cu ft)	Mean Temp F	THICKNESS OF INSULATION—INCHES														
				1/3	3/8	7/16	1/2	2/3	3/4	7/8	1	1 1/2	1 5/8	1 3/4	2	3	3 5/8	5 5/8
Balsam Wool	Chemically treated wood fiber	2.2	90				0.54				0.27							
Cabots Quilt	Eel grass between Kraft paper	4.6	90	0.78			0.52	0.39			0.26							
Cabots Quilt	Eel grass between Kraft paper	3.4	90	0.75			0.50	0.375			0.25							
Celotex	Board form insulation made from sugar cane fiber	13.2	90			0.777	0.68			0.389	0.34	0.227		0.194	0.17	0.113		
Corkboard	Pure; no added binder	14.0	90								0.34	0.227			0.17	0.113		
Corkboard	Pure; no added binder	10.6	90								0.30	0.20			0.15	0.10		
Corkboard	Pure; no added binder	7.0	90								0.27	0.18			0.135	0.09		
Corkboard	Pure; no added binder	5.4	90								0.25	0.167			0.125	0.083		
Corkboard (Eureka)	Asphaltic binder	14.5	90								0.32	0.213			0.16	0.107		
Dry Zero	Kapok between burlap or paper	2.0	90								0.25							
Dry Zero	Kapok between burlap or paper	1.0	90								0.24							
Fibrofelt	Flax and rye fiber	13.6	90	0.96	0.853		0.64		0.426		0.32							
Flaxlinum	Flax fiber	13.0	90				0.62				0.31							
Hairinsul	(75 per cent hair) (25 per cent jute)	6.3	90				0.54		0.36		0.27							

Hairinsul	(50 per cent hair) (50 per cent jute)	6.1	90				0.52		0.347		0.26							
Hair felt	Felted Cattle Hair	13.0	90				0.52				0.26							
Hair felt	Felted Cattle Hair	11.0	90				0.52				0.26							
Insulex or Pyrocell	Cellular Gypsum	30.0	90								1.00		0.615				0.276	0.178
Insulex or Pyrocell	Cellular Gypsum	24.0	90								0.77		0.473				0.212	0.137
Insulex or Pyrocell	Cellular Gypsum	18.0	90								0.59		0.363				0.163	0.105
Insulex or Pyrocell	Cellular Gypsum	12.0	90								0.44		0.271				0.121	0.078
Insulite	Board form insulation made from wood	16.9	90				0.68				0.34							
Linofelt	Flax fibers between paper	4.9	90		0.747		0.56				0.28							
Lith	Rock wool, flax and straw pulp with binder	14.3	90								0.40	0.267			0.20	0.133		
Regranulated Cork	About 3/16 in. particles	8.1	90								0.31		0.191				0.086	0.055
Rock Cork	Rock wool block with binder	16.7	86								0.37				0.185	0.123		
Rock Wool	Fibrous material, made from rock	10.0	90								0.27		0.166				0.075	0.048
Sprayo-Flake	Shredded paper with silica binder	4.2	94				0.56		0.373	0.32	0.28	0.186			0.14	0.093		
Thermofelt	Jute and asbestos fibers, felted	10.0	90				0.74				0.37							
Thermofelt	Hair and asbestos fibers, felted	7.8	90				0.56				0.28							
Thermofill	Dry, Fluffy, Flaked Gypsum	34.0	90								0.60		0.369				0.165	0.107
Thermofill	Dry, Fluffy, Flaked Gypsum	26.0	90								0.52		0.32				0.143	0.092
Torfoleum	Peat Moss compressed into sheet form	10.2	91.5								0.29	0.193		0.166	0.145	0.097		

Theoretically, a *perfect* insulation would reduce the heat transmission through a wall or roof to zero, and would, therefore, be classified as a 100 per cent insulation. The degree to which a 100 per cent insulated construction is obtained by any given thickness of a material of a certain conductivity would govern the classification of the material. Suppose, for example, the uninsulated construction were a frame wall, consisting of wood siding, building paper, wood sheathing, studding, wood lath and plaster, having a coefficient of transmission of 0.227 Btu per hour per square foot per degree Fahrenheit. (Table 8-A, p. 33, Chapter I, the A. S. H. & V. E. GUIDE, 1929). If the sheathing, building paper and lath of this construction are replaced with two ½ in. thicknesses of a rigid insulation, the heat transmission will be reduced to 0.157 Btu per hour per square foot per degree Fahrenheit, and the value of E for this material in this case would be—

$$100 \times \frac{0.227 - 0.157}{0.227} = 31 \text{ per cent.}^{3}$$

If, instead of the rigid insulation being used in place of the sheathing and lath, a 1-in. thickness of a flexible insulation, having a conductivity of 0.27 Btu per hour per square foot per degree Fahrenheit per inch of thickness, is installed between the studding, the coefficient of heat transmission will be 0.104 Btu per hour per square foot per degree Fahrenheit per inch of thickness, and E for the thickness of this insulation installed, for the manner of installation and for the construction, will be—

$$100 \times \frac{0.227 - 0.104}{0.227} = 54.3 \text{ per cent.}^{3}$$

If the spaces between studding are filled with a cellular gypsum fill, the heat transmission coefficient will be 0.110 Btu per hour per square foot per degree Fahrenheit, and the efficiency of the insulation or the value of E in this case will be—

$$100 \times \frac{(0.227 - 0.110)}{0.227} = 51.5 \text{ per cent.}^{3}$$

By this method, insulations could be rated on a percentage basis or classified in a manner similar to that shown in Table 5.

Recent experiments conducted at the University of Illinois show that as the heat transmission coefficient of a ceiling is decreased by the addition of insulation, the temperature of the air at the ceiling is increased and the temperature above the ceiling is decreased, thus tending to offset, in a measure, the effect of the insulation. Hence, although one insulation may reduce the heat transmission through a ceiling or a roof twice as much as another, it will not be twice as effective because of the increase in the heat head or temperature difference. Therefore, accurate comparisons involving overall transmissions of constructions should take account of this fact, the importance of which cannot be overemphasized.

RATING ON BASIS OF COST PER UNIT OF RESISTANCE

Thus far no consideration has been given in this paper to costs. Heating

[3] The value of E is not an index of the percentage saved of the fuel or radiation required to heat a building. The percentage saved of the total heat losses obtained by insulating the walls and/or roof of a building must necessarily involve all of the heat losses, including the infiltration losses and the transmission losses through glass.

TABLE 4—CONDUCTANCES OF VARIOUS THICKNESSES OF INSULATIONS BASED ON TESTS CONDUCTED AT ARMOUR INSTITUTE BY PROF. J. C. PEEBLES AND COMPUTED FROM CONDUCTIVITIES PER INCH OF THICKNESS

Material	Description	Density (lb per cu ft)	Mean Temp of Sample F	THICKNESS OF INSULATION—INCHES									
				7/16	1/2	7/8	1	1 1/2	1 5/8	2	3	3 5/8	5 5/8
Celotex	Board form insulation made from sugar cane fiber	13.5	70	0.754	0.66	0.377	0.33	0.22		0.165	0.11		
Flaxlinum	Flax fiber	14.0	70		0.64		0.32						
Homasote	Wall board (5/16" thick made from paper pulp)	22.9	70				0.40						
Inso Board	Board form insulation made from wheat straw	17.0	68	0.754			0.33						
Insulite	Board form insulation made from wood pulp	16.5	70		0.68		0.34						
Keystone Hair	Hairfelt between layers of paper; 1/4" thick	11.0	75				0.25						
Lith	Rock wool, flax and straw pulp with binder	14.5	75				0.38	0.253		0.19	0.127		
Masonite	Board form insulation made from exploded wood fiber	18.0	75	0.754			0.33						
Maftex	Board form insulation made from roots of licorice	16.1	81	0.77			0.337						
Pyrocell or Insulex	Cellular Gypsum	30.0	75				0.92		0.566			0.254	0.164
Pyrocell or Insulex	Cellular Gypsum	24.0	75				0.737		0.453			0.203	0.131
Pyrocell or Insulex	Cellular Gypsum	18.0	75				0.566		0.348			0.156	0.10
Pyrocell or Insulex	Cellular Gypsum	12.0	75				0.400		0.246			0.110	0.071
Sprayo-Flake	Shredded paper with silica binder	4.5			0.50	0.286	0.25	0.167		0.125			
Thermofill	Dry, fluffy, flaked gypsum	24.0	75				0.475		0.292			0.131	0.084
Thermofill	Dry, fluffy, flaked gypsum	18.0	75				0.34		0.209			0.094	0.06
Thermosote Insulating Board	Board form insulation made from wood pulp	20.8	70	0.854			0.374						

engineers are usually concerned with the matter of insulation primarily from the standpoint of the proper design and functioning of the heating system. But whether the engineer or some one else is the deciding factor in the selection of the insulation, an intelligent comparison cannot be made unless some consideration is given to costs, assuming that all materials involved are suitable for building purposes.

Neglecting all other factors but cost of insulation, thickness and the conductivity, it is possible to obtain a comparatively simple basis for comparison, namely, the cost per unit of resistance. For example, a so-called rigid insulation, such as cork board, having a conductivity of 0.30 per in., and costing 15c per sq ft per 1½ in. thickness, would cost $\frac{0.30 \times 15}{1.50}$ or 3c per sq ft per unit of resistance. A ½ in. thickness of a flexible insulation having a conductivity of 0.27 Btu per hour per square foot per degree Fahrenheit per inch of thickness, and costing 4½c per sq ft would cost $\frac{0.27 \times 45}{0.50}$ or 2.43c per sq ft per unit of resistance. A fill having a conductivity of 0.50 per in. and costing 16c per sq ft for a thickness of 3⅝ in. would cost $\frac{0.50 \times 16}{3.625}$ or 2.2c per square foot per unit of resistance.

TABLE 5—CLASSIFICATION OF INSULATIONS ON BASIS OF EFFICIENCY OF INSULATION FOR ANY SPECIFIC TYPE OF CONSTRUCTION

RATING	$E = \frac{100 \times (U_o - U_i)}{U_o}$
Class A	90% to 100%
Class B	80% to 90%
Class C	70% to 80%
Class D	60% to 70%
Class E	50% to 60%
Class F	40% to 50%
Class G	30% to 40%
Class H	20% to 30%
Class I	10% to 20%

A more accurate scheme for rating insulations, according to the cost per unit of resistance, would be on an *installed* basis, by which consideration would be given to the cost of installation, as well as the effective resistance of the insulation as applied, thus allowing for any increase or decrease in the number of air spaces in the construction, and for any materials replaced by the insulation. In many cases the cost on an "installed" basis would be comparable to the cost on the material alone basis, but in other cases there would undoubtedly be considerable difference.

RATING ON BASIS OF ECONOMIC VALUE

One insulation may reduce the heat transmission of a certain construction to a much greater extent than another, but if the cost of the one greatly exceeds that of the other, the monetary value of the additional heat saving of the more expensive insulation may not be sufficient to justify its use; in fact, the more expensive insulation may actually be an extravagance when considered from the standpoint of return on the additional investment required.

To further illustrate this point, suppose the heat transmission of a certain uninsulated wall to be 0.25 Btu per hour per square foot per degree Fahrenheit, and that one type of insulation installed in the thickness in which it is usually applied will reduce the heat transmission to 0.15 Btu per hour, per square foot, per degree Fahrenheit, and another type will reduce it to 0.12 Btu per hour per square foot per degree Fahrenheit. The fuel saving in the former case is estimated to be 1.26[4] tons of coal per 1,000 sq ft of wall area per heating season, and would have a value of $12.60 per year based on coal at $10.00 per ton. The fuel saving in the case of the material which will reduce the heat transmission to 0.12, is estimated on the same basis to be 1.64[4] tons of coal per heating season, and would have an annual value of $16.40 if the price of coal is $10.00 per ton. If the costs of the two insulations as installed are $100.00 and $200.00, respectively, per 1,000 sq ft of wall area, the former will result in a return on the investment of 12.6 per cent, and the latter 8.2 per cent, disregarding radiation saving, depreciation, etc. The more costly insulation in this case will save only 0.38 tons of coal per heating season more than the less expensive insulation, representing an additional monetary return of only $3.80 per heating season at an additional cost of $100.00. The return on the investment of the more expensive insulation is therefore only 3.8 per cent, as compared with the first, and from the economic standpoint, would not justify the use of the more expensive insulation in this case.

Obviously, the more costly the fuel, the greater the thickness of insulation required to obtain the proper economic balance. For gas and electricity, a well insulated wall is an economic necessity. For less expensive fuels, however, more insulation than will result in an adequate annual return on the investment is a waste of money, if in determining this return all of the tangible savings accruing from the use of the insulation are taken into consideration. This does not mean that the higher the return on the investment, the better the material, because the initial thickness of any given insulation will produce a higher return on the investment than any equal subsequent thickness. In other words, each increment of insulation applied, will result in a smaller return on the investment than the preceding increment, assuming that the installed cost of each additional increment is the same. This is sometimes called the law of diminishing return. Although the return diminishes with each unit of thickness of insulation added, several thicknesses of a given form of insulation may be warranted for a certain type of construction. Of course, few forms of insulation are not susceptible of application in progressive thicknesses.

There is, however, a practical economic limit to the amount of insulation that should be installed for any given type of construction, and for any given set of conditions. Hence, in order to classify insulations according to their

[4] Based on a heating season of 5040 hours, an average temperature difference of 30 F, 12,000 Btu per pound of coal and an overall efficiency of the heating system of 50 per cent.

economic values, it would be necessary to consider the type of construction involved in each case, the conductivities, costs and thicknesses of materials, fuel and radiation savings and their monetary values, depreciation and other factors. It seems apparent, therefore, that the problem of properly classifying insulations from the economic standpoint and of determining the optimum thickness, becomes too involved to lend itself to a simple solution.

CONCLUSIONS

There is an apparent need for a method of appraising or evaluating insulations. To establish a fair and equitable basis for comparison, however, does not appear to be an easy matter. In order to encourage further thought on the subject, five plans are suggested. The first is based solely on conductivities per inch of thickness. The second is a resistance method taking into account the thickness in which the insulation is installed. The third plan is intended to rate insulations according to the extent to which they reduce the rate of heat transmission through a wall or roof, and, therefore, involves the type of construction, the thickness of insulation and the manner of installation. The fourth scheme would rate insulations according to the cost per unit of resistance, either for the material alone or on an installed basis. The last plan suggests the possibility of appraising insulations according to the return on the investment.

It is evident that this subject is one which warrants further study for the purpose of rationalizing the rating and classification of insulations and eliminating quibbling over small, insignificant differences in the conductivities, which is so detrimental to the progress of the insulation industry.

DISCUSSION

H. B. LINDSAY (WRITTEN): I think it is probable that my appreciation of the earnest and practically expressed plea for a competent method of appraising insulations, so well advanced by Mr. Close, is shared by all who have heard it. It is only my regret that I shall not have the opportunity of hearing it with the added force of the earnest personality back of it.

In the following comments I would ask you to bear in mind that what research I have been able to do, has been in the low temperature but high duty field where insulation is—or should be—supreme, and every tenth of a Btu per hour may make an overall difference of many hundreds of thousands of dollars a year.

As Mr. Close very properly points out, in the insulation of residential buildings we need not be so meticulous, for we have both a considerable degree of insulation effect present in the uninsulated wall itself, and a comparatively large and unavoidable heat leakage in any event through windows, doors, ventilation and through the structural material. These conditions are largely the opposite to those of refrigerative work.

This situation renders the economic limits of insulation in the two fields, though alike in principle, so utterly divergent in degree, that I do not believe the same basis of evaluation can reasonably be used for both. I would therefore respectfully suggest that the whole matter might well be indicated as referring to the ordinary building field rather than the refrigerative.

Before briefly discussing the possible difficulties involved in attaining the end designed in this paper, permit me to strongly compliment Mr. Close on his gently caustic references to ridiculous insulating claims that are all too often made for materials of many sorts, claims not remotely related to fact. As one who has worked a good many years to arrive closer to the *facts* of insulation, I could wish Mr. Close had dealt more caustically and less gently with such perverters of—or at least triflers with—fact.

Mr. Close indicates five suggested methods for classifying insulants. He rightly points out that more factors than just insulating efficiency should be considered in such a classification. Indeed, I suppose that the ultimate nature of this classification should be such as to enable an Architect to select the most suitable insulants for a given building with given structure and under given conditions.

But what does this mean? Essentially it means indicating those materials which will procure the greatest possible reduction of total or overall heat transmission with the least possible investment. To reach this desirable mean, however, involves more factors than insulation value and cost. We are considering an insulation intended to retain its efficiency for a long period of years. We must therefore also look for permanence, and this in turn is affected by both inherent and climatic conditions.

With such factors included, then as far as essential method of rating goes, I for one am willing to agree with Mr. Close that the most desirable probably lies with his suggested methods 4 or 5. It seems to me, as a matter of fact, that method 5 is simply method 4 with the economic limit introduced. Is it necessary to introduce this? Won't that factor vary to some extent with locality and individual? In any event, under method 4, cannot the Engineer, the Architect, or even the Owner (if he is in the picture) figure out his economic limit if he has the ratings of the various suitable insulations and the *cost per unit of resistance?* This cost, however, should include the normal cost of application, and, in the case of the board insulations, their total resistances for successive thicknesses, both when installed in contact or with intervening air spaces.

Such a table I believe would have value, but I believe that it can only be made usefully accurate by much and patient testing *under the actual conditions of use*. This is undoubtedly an admission that in the complete formulating of our science in insulation, we have a long way to go. But it is sadly true. Let me give you a few instances of this which indicate the necessity of such tests to be made under the condition of use.

We speak of air spaces between sheets of insulation or between them and the inner faces of the wall. If horizontal these, of course, have a marked effect when heat is endeavoring to penetrate downward. But in the side wall or sloping roof, the refrigerating engineer at present allows an hourly resistance of 0.50 for each surface of material on the two sides of that air space—*provided* it is ½ in. or more, or 1 in. or more, as his training may indicate. He does not consider the nature of the surfaces exposed to that air.

Along the same thought, when Dr. Lisse of the Sprengluftgesellschaft of Berlin came to see how we had so successfully insulated the liquid oxygen soaking chests for their interesting explosive in this country, he told me they were using in Germany sheets of metal foil for insulation.

As applied to the possible table under discussion, the chances are that the

classification of the various exposed insulation surfaces (where air spaces were involved) would be the same and the variation slight—but we should make doubly sure by proper test.

Again, for such a table of classifications, the actual effect of increasing thicknesses of the same material should be determined by test. I will suggest that the curve of resistance for the increasing thicknesses (excluding any external resistance) will be found parabolic rather than straight, as we all usually assume in our computations. The actual deflection of this curve from the straight curve for resistance when considered exactly proportional to thickness, may well prove sufficient to throw a certain thickness of any insulation out of expected line.

There is the factor of heat passage through the structure of a wood frame building. With 2 x 4 studs spaced 16 in. centers, not to mention sills, plates, braces, etc., with a conductivity of about 0.3 (allowing for considerable moisture content), one sees at once that under some conditions of application this becomes a levelling factor between different efficiencies of the insulation, while in other applications it is negligible.

Another factor is the upsetting effect of moisture absorption. We are prone to calculate our transmissions on the basis of conductivities ascertained in a nearly bone dry condition, a condition never obtained during construction or subsequently. With an outside temperature of 0 F and inside of 70 F, particularly when the inside air is conditioned, the walls and their insulation will certainly acquire considerable moisture content in time. Prof. L. F. Miller of the University of Minnesota has very ably shown the effect of varying moisture content on the conductivities of a number of the more used insulations. He shows that with the additional absorption due to a change of only 10 per cent in relative humidity of the involving air (taken, for example, at 70 F) conductivities of the materials shown increase from 2 per cent to 2½ per cent. This is, of course, all below the dew-point temperature. The actual establishment of dew-points in insulated house walls may not be entirely far-fetched in view of the general lack of any real moisture seal on the inner side of the insulation space.

I may possibly seem to be too particular in digging up these various complexing factors and variants. But we are discussing a method of appraisal of materials for uses which undoubtedly involve these details. Such an appraisal or classification would be useless—nay perhaps an unintentional injury, unless it was accurate for actual use. Even so, two materials might actually exchange classifications under markedly different climatic conditions.

The point I would most respectfully urge is that in undertaking such a classification we are contending with a number of varying conditions which do not obtain in the naturally more exact practice of refrigerative insulation. And that the safe method of proceeding is by careful and competent tests under the actual conditions of use, just as far as that is practicable.

H. J. SCHWEIM (WRITTEN): This paper is greatly needed in the Industry at this time. The world is *Insulation conscious*. It is, however, only conscious of the fact that houses should be insulated. How to appraise the different types and kinds of insulation is unknown to them, and the advertising of insulation manufacturers only tends to confuse them.

What the home owner is interested in, is what percentage of reduction of

heat loss through the wall the various types of insulation will provide. Plan No. 4 suggestion by Mr. Close gives the cost per square foot per unit of resistance, but to a layman what does *unit of resistance mean.* Also this plan does not take into consideration the manner in which the insulation is used. Is it substituted for some other material that has insulation value, is it placed in the center of the wall so as to provide two additional surface factors, or does it fill the entire space and cut out two surface factors? I do not see how insulating materials can be compared except on the basis of transmission through the completed wall.

There is no direct relationship between the conductivity per inch of thickness of an insulating material, and the heat loss through the wall in which that insulating material is used. By that I mean simply because the conductivity of one material is one half that of another it does not follow that the heat loss through the wall is twice as much in one case as in the other. Still, when comparison between two insulating materials made in the public press are based on the conductivities of these materials, the layman or home-owner obtains the impression that the heat loss through walls in which these materials are used will vary as two to one. Nothing is further from the truth. When comparisons are made based upon the hot plate test, the conductivities are given per inch of thickness regardless of the commercial thickness of the respective materials, and no consideration is given to the manner of installation. As an example, assume that in Wall *A* an insulating board ½ in. thick with a conductivity of 0.30 per inch is used as sheathing, and in Wall *B* a 1 in. thick board with a conductivity of 0.60 per inch is used in the center of the wall between the studs.

Based on conductivities the insulating board used in Wall *A* is twice as efficient as the one used in Wall *B*. However, due to the difference in thickness of the board, and in the manner in which they are used, the coefficient of transmission through Wall *A* is 0.19, and through Wall *B* is 0.132. In other words the insulating material that is one half as efficient as the other based on the difference between their conductivities is in reality 50 per cent better when considering the heat loss through the wall, or if these two insulating materials are appraised in accordance with the third plan suggested by Mr. Close the insulation material used in Wall *A* is but 16 per cent efficient, while that used in Wall *B* is 41 per cent. efficient. My contention is that advertising listing competitive insulations showing conductivity per inch of respective materials regardless of the commercial thickness, or manner in which used gives the consumer an erroneous impression of the respective values of the materials advertised. Conductivity determined by the hot plate method is not a true measure of the insulation as surface coefficients, infiltration, etc., are ignored. Comparisons in my opinion must be based on the relative transmission through the completed wall, as suggested by method 4 of this paper.

J. H. Bracken: I think a little reflection on the arguments advanced in this paper will show that the difficulties are sales difficulties; that some of the manufacturers who speak about advertising by other manufacturers and their claims of value, are seeking some easy method whereby they may be spared the necessary expense of telling the public what their own sales qualities are.

The manufacturer who complains about the other manufacturer has a golden opportunity of setting himself right with the public. All he has to do is to

spend an equivalent sum of money for advertising and to hire an equivalent number of salesmen and devote his attention to telling the public through these channels what his own insulation can do. I don't think we should forget this point, because it is behind, in my opinion, this entire discussion.

I should like to say a word about the hot plate test, although you understand that it needs no defense from me. The hot plate test is a method—the best one we know, I think—for determining the conductivity of a homogeneous material. The results obtained of necessity have to be expressed in a standard thickness, a standard area, under standard conditions, else no comparison is possible. That test has been adopted not only by this Society, but by the *Bureau of Standards,* the *National Research Council,* the *American Society of Refrigerating Engineers,* and I think without exception by every technical school in the United States. If that hot plate test is too grossly material, so to speak, for some of these insulating materials, because it does not reveal the hidden spiritual qualities that some of them have, it is the duty of the manufacturers of those materials to explain such spiritual qualities at their own expense.

This is a valuable paper for the personal study of an engineer since it suggests various factors that may be considered in the purchase, for a particular use, of a particular insulating material but I am much opposed to the principle on which the paper is based and I am of the opinion that it tends to aggravate the malady it aims to cure, namely to prevent ordinary building materials from masquerading as insulations.

I think this Society may well say that an insulation shall not possess a conductivity greater than 50 per cent of that of lumber for an equivalent thickness. Such a definition would admit all of the manufactured insulations and exclude claims of old well known building materials which are advancing insulation arguments because of large insulation advertising and for no other reason.

Some such result, it seems to me, is what this paper set out to achieve since the introduction definitely states that the purpose of the paper is to help to differentiate between insulating materials and those not entitled to this classification; further, that there are controversies over small differences in conductivities of insulation which are usually within the error of testing. The proof, however, shoots far beyond this prospectus. In order to prevent common building materials from calling themselves insulations, the author would precipitate a war among insulation manufacturers. To prevent the rebels from winning he would kill the loyalists. For you may be certain that the publication by this Society of Table 1, elevating certain insulations into a preferred Class *A* will be followed by serious consequences, and in my opinion serious injustice will be done to the insulation industry.

An insulation is not a simple thing like a metal which can set in a table of conductives but is a very complex material. Its conductivity value is only one quality it possesses and that may not be the most important quality for many uses but may be over-balanced by strength, or sizes of pieces, or availability, or appearance, or capacity for easy handling, or capacity for ornamentation, or for other reasons and qualities. You may as well endeavor to classify hats or overcoats on the basis of weight, putting a carters coat into Class *A* and a rain coat into Class *F*.

The harm resulting from the publication of such a table is manifest. Outside of this room and away from the presence of men who understand that Table *A*

relates to conductivities only, it will be said that this Society has conferred the D. S. C. on Class *A* and all other insulations are inferior.

You cannot escape that and I do not think you have the right to do it. This Class *A* relates to three of the five suggested methods of appraising insulations because conductivity, resistance, and conductance are intimately related. What are we to say about the fourth and fifth suggested methods of appraisal in which the factor of price is added to conductivity and an economic value of the return on the investment introduced so that insulations in Class *A* of the first method may be found in Class *D* or Class *F* of the fourth and fifth methods.

To sum up, I am opposed wholly to the principle of this paper which seeks to establish for the engineer a buying rule of thumb to direct him in the specification or purchase of these materials. THE GUIDE provides for the engineer all the data shown in this paper and the manufacturers may be trusted to present costs and sales qualities and necessary sales services and they should be let strictly alone in these activities.

The principle to which I object in this paper is that it introduces buying advice, or commercial practice into a technical discussion by inference at least. What this Society should seriously set out to do is what this paper purported to do, namely, set down a clear definition which will distinguish insulating materials from those materials which falsely claim to have insulating value.

J. D. CASSELL: The last paper covered the point very well. I do not think it is the function of this Society or its Research Bureau to go into testing the material in a building, because I may install in one way that would be detrimental, and my competitor install in another way that would be beneficial, we cannot reach out that far. If we define the densities—if that enters into it; I am not technical enough to understand that—if we define the densities of each product as manufactured, and then the resistance through that product or to that product, I think our Research Bureau has gone far enough. We cannot hope to control installation, nor can we hope to control manufacture, other than by, as I said before, the density of the product and its insulation qualities.

F. B. ROWLEY: Mr. Chairman, I would like to say just a word about this paper. It seems to me there is no question in most of our minds that there is some confusion in the methods of rating insulating materials. To the man who is not familiar with the units in which conductivities are expressed, it is rather confusing to compare materials of different conductivity and of different thickness. The purchaser is often allowed to overlook the fact that materials are rated on the basis of one inch in thickness, and not on the thickness as sold.

While there are plenty of data available for the man who is familiar with the insulating materials, to make comparisons, I believe that for the average man, it would be less confusing if materials were rated either on the conductance or total heat resistance as sold. This would give the purchaser a measure of the actual insulation he is buying.

If materials were to be rated on the basis of *A, B, C,* etc., the problem would then be to define the values of these ratings. The values might be made up on a heat resisting basis, in which case an arbitrary line would have to be drawn between the different grades. There is no logical reason why a material with a conductivity of 0.30, for instance, should be placed in Class *A* while one with 0.32 as its value might be placed in Class *B*. The second material might on the whole be of a superior quality, yet in the public mind it would be rated

as inferior. As an alternative, let us suppose that other qualities are also considered. Which ones will be counted and to what extent? Insulating materials have qualities of strength, rigidity, flexibility, porosity, ability to withstand water, resistance to deterioration, economic value, etc. How many of these qualities and to what extent should they be considered in making a classification? Each of these qualities have definite measuring units which are more or less clear in the public mind. It seems as if it would be a mistake to group them under a single classification or to try to devise a new single standard which would require a new definition. It is doubtful whether or not any rating could be devised which would meet with anything like unanimous approval. The problem is too complicated and has too many variable factors. An attempted solution might make the condition worse and not better. Such an arbitrary or combination unit might be made up to fit one main requirement, but it could not serve all requirements. I think that the paper has done what its author probably intended for it to do; to stir up some discussion on this question, and to bring out some of the points where we are at a difference on the rating of materials.

HOMER R. LINN: I think we should give Mr. Close a vote of thanks for at least calling attention to the value of building insulation. I am frank in saying I question whether this paper will get us anywhere, because personally I do not see how any of his classifications can meet with the approval of the insulating manufacturers and unless the work done along this line is done in conjunction with the insulating manufacturers, it will get no further than did the Boiler Testing and Rating Code Committee until they condescended to co-operate with boiler manufacturers.

A couple of years ago I had the pleasure of serving on the Insulating Committee of the Illinois Master Plumbers' Association with Prof. Kratz and two other gentlemen. We did not attempt to determine the conductivity of the different insulating materials. However, we did put them in three classes to serve our purpose in making a study of the effect of building insulation on the heating system. We found that the same insulating material installed in different ways had an entirely different effect on the heat saved. For instance, the manner in which this insulation was installed had a greater effect in most cases than did the conductivity per inch. I am inclined to think that the per inch thickness value probably is not the right value for the man to consider who is buying insulation.

The Illinois Master Plumbers have published a pamphlet giving a suggested method for figuring the heating system of a house insulated in a number of different ways. This Bulletin has been widely distributed by them. Mr. Cassell wisely remarks that he is interested in the effect of what insulation does to the heating of his building rather than what classification the insulation falls in.

I really think that something should be done in an earnest endeavor by this Society in the study of a proposed method of the valuation of insulation as it affects the heating system. I would like to suggest that a Committee be appointed to make such a study.

A. P. KRATZ: It seems to me the whole matter is more or less one of sales psychology. To the engineer there is no confusion and it is a matter of indifference; so long as he has the fundamental constants he can make his calculations and base his conclusions on whatever the results of his calculations are,

and make his comparison on that basis. When it comes to the psychological effect on the general public, I do not know that I am competent to draw any conclusions.

J. D. Shodron: As a buyer of insulation, about 75 carloads a year, or from three to five million square feet per year, we are interested in another factor that has not been mentioned here, and that is, infiltration, or air in-leakage and out-leakage through an insulation. We find in our work where insulation is used in wall construction that the porosity of the material is a very important factor, and I would like to hear from Mr. Close whether that has been given any consideration. We have found some of the insulators to pass from one and one-half cubic feet of air per hour per square foot, to as high as 35 cu ft per hour per square foot, insulators having about the same Btu transmission per square foot per inch thickness.

Now, if an insulation is made with a very dense surface on one side, it will reduce the amount of air in-leakage and thereby effect a considerable saving in heat. You can liken it to putting on a sweater and going out and facing a cold wind, and you will feel cold, whereas if you put on a light-weight, water-proof, air-tight raincoat, you could go out and face the same wind and be really comfortable. If you could get the combination of the two you would have the benefit of both factors. I would, therefore, like to see that also be given some consideration when establishing the value of an insulation.

G. T. Pearce: I might say in regard to the matter of insulation or application of insulation that in the more definite field of insulation, refrigeration, we always make sure that the insulation is put in properly whenever we sell a customer. Now, whether it is feasible to do that through the general public, through distributors, I don't know. We control all of our distribution points.

Another feature that comes up is the matter of air infiltration through insulation. In our case we absolutely insist that a definite, fixed, and tight air-proof and moisture-proof seal is a part of the installation of the insulation, and that probably has brought up this question. There is not a building paper on the market today that in itself as an insulation is of any value. It may be so installed that it will set up a surface resistance to the passage of heat, but the material itself is of such thickness and such conductivity that in itself it can be absolutely ignored.

L. A. Harding: It would be difficult if not altogether impossible to classify insulators on the basis of economic performance. The varying material and labor cost for installation in different parts of the country as well as the varying cost of fuel might show that particular type of insulation to be economical one place, while another insulator would be the most economical one to employ in another place.

I believe that conductivity and surface coefficient tables are all that is required. The economic problem is one for the engineers to solve in any specific case and not the owner or manufacturer of the article. The problem is simply this— Will this saving (fuel) due to the installation pay a sufficient return on the added investment required to make it worth while? This will depend on what the owner may require as to the return on his investment. Manufacturers when considering savings due to recovery of wastes (heat or materials) usually insist on a 15 to 20 per cent return.

PRESIDENT LEWIS: It might be pointed out in this connection that if the Society were to undertake to classify insulation on the basis of its total installed cost with reference to its efficiency, the boiler manufacturers might be the next to come along and ask us to do that, and if so, we might as well disband the Society.

PAUL D. CLOSE: As I anticipated, there seem to be two sides to the story, those for and those against. It is my opinion that an insulation rating code is needed just as rating codes are needed for other materials and equipment used in the heating industry. Of course, from the strictly engineering or scientific standpoint, a rating code should not take into consideration costs in the method of rating, but in the selection of an insulation, cost of course is a factor. The opinions expressed in most cases were undoubtedly honest convictions, but it is not unlikely that in some cases the opinions expressed were influenced by how the individual might be affected by a method of rating.

Mr. Schweim took exception to the fourth method suggested involving the unit of resistance, on account of the fact that the average person would not know what was meant by *unit of resistance*. I believe that also applies to *all* the phraseology used in connection with insulation. The average person does not know what it means. He also took exception to the fourth method on account of the fact that it does not take into consideration the manner in which the insulation is installed but that is taken care of in one of the other suggested methods.

Mr. Bracken uses an analogy in connection with hats and articles of wearing apparel, stating that one might have some difficulty in appraising them, and that it would be just as logical to attempt to appraise insulations or rate them. Although it is true that there are certain things that are of such an intangible nature that it is difficult to rate them, on the other hand, there are some commodities that very definitely are susceptible of being rated. The Society has already established a boiler rating code, and others are contemplated.

In the matter of infiltration, there is undoubtedly a considerable difference in the infiltration through various materials, but this is of minor importance when other things are considered. For example, the claim is being made that the infiltration through a certain material is ten times greater than through another. An analysis of these figures indicates that such comparisons are in most cases ridiculous; it is like saying that a match cost ten times more than a toothpick. Ten times more is a high ratio, but a match itself does not cost very much, so you can forget about both of them. Furthermore, even though a material is more or less porous when it is installed in a wall with the other materials, particularly building paper or plaster, the infiltration usually is a negligible quantity.

Mr. Harding intimated that the method of rating insulations on the basis of return on the investment was more or less complicated, and I am inclined to agree with him. If one endeavored to take into consideration every conceivable factor that might enter into the problem, it would become hopelessly complicated.

No. 846

TIME LAG AS A FACTOR IN HEATING ENGINEERING PRACTICE

By JAMES GOVAN[1], TORONTO, CANADA

NON-MEMBER

THE chief purpose of this paper is to suggest an alternative definition of *Resistance* as used in determining heat transmission coefficients.

Where the definition of Resistance as used is as follows:

The Resistance, *R*, of material arranged in layers each of uniform thickness between parallel faces is equal to the number of degrees Fahrenheit difference in temperature of the faces required to maintain a rate of heat flow of 1 Btu per hour per square foot of area; the suggested alternative definition of Resistance would be:

The Resistance, *R*, of material arranged in layers each of uniform thickness between parallel faces is equal to the number of hours required for 1 Btu to flow through 1 sq ft of area, when there is a difference of 1 F in temperature between the faces.

Similarly, Surface Resistance would be expressed in hours for 1 Btu to flow through one square foot of surface, when there is 1 F difference between a fluid medium and the surface with which the medium is in contact.

With the adoption of the suggested alternative definition of Resistance, there would follow a clearer perception of walls and roofs of buildings as blankets delaying, more or less effectively, the passage of heat through them, in addition to performing their more generally understood functions as protectors against wind, rain and snow, and providing privacy, carrying loads, etc.

The secondary purpose of the paper is to call attention to the effect of the retention of heat in buildings having very greatly increased resistance to heat flow, when the calculations to determine the size of the heating plant are based on the recommendations of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS' GUIDE.

The thought prompting the presentation of the paper by an architect is the need for a better understanding between engineer and architect of the mutual problems affecting not only themselves, but, more so, their clients.

[1] Member Royal Architectural Institute of Canada.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Bigwin Inn, Lake-of-Bays, Ontario, Canada, June, 1929.

Reasons for Suggested Alternative Definition

The mere fact that you have been so gracious as to invite an architect to present such a paper as this, indicates that some, at least, of the members of your Society are willing to study your work from the outsider's angle, and that all of them don't feel that you have to make your engineering data more difficult to understand in order to preserve for your profession the privileges of heating engineering practice.

If by any chance your ranks are divided, like those of the medical profession, into two groups, one anxious to reduce the cost of service to the lowest possible point, and the other resisting to the utmost any departure from the recognized and approved professional position, let me assure the latter that the more the public (and that includes architects) clearly understand the elementary factors that govern your work, the more they will be willing to allow themselves to be guided by your members and less by unscientific individuals, who happen to know where to buy, on credit, boilers, pipe and fittings.

If it be conceded then that there would be an advantage to the engineer, to the architect and to the public if there was a closer understanding between the engineer and architect, as to the service that can only be rendered by a competent engineer, the first necessity, in the creation of such an understanding, is that both parties should use language that both can understand.

Now architects may be uneducated or pig-headed or lazy or conceited or they may be all four at once, but the fact remains that very few of them can or will take the trouble to understand even the definitions of the terms that you use in your studies of their buildings, to determine whether their walls, roofs and openings will leak heat like sieves or give their clients reasonable service for money expended.

That this difficulty besets others than architects can be clearly demonstrated, if one tries to address a nontechnical audience on the subject of heat losses from buildings, because, under these circumstances, it becomes at once apparent that any attempt to use the language of the official GUIDE produces such confusion of thought, as is evidenced every day in the advertising of the various insulating materials now spread before us, not only in our technical journals but also in the daily press and even more so in the popular magazines.

The modification of your accepted definition of Resistance, that I am suggesting for your consideration, is one that I was prompted to make several years ago to a small group of business associates, and its reception then, and since at a meeting of the *Better Business Bureau* in New York over a year ago and at a meeting of the Toronto Branch of the *American Society of Mechanical Engineers,* indicates that it had something of novelty and sufficient merit to provoke favorable comment.

Whether your members may have considered the idea before I know not, but THE GUIDE 1929 continues to use the accepted definition, so I presume that so far as your own needs are concerned the official position is that there is no occasion for change.

As in the case of the doctor's Latin prescription, this attitude may serve a useful purpose between the professional engineer, who detects the defects in a building and prescribes accordingly, and his contractor colleague, who fills the prescription for the architect and his client to swallow. But when the broad-

casting of advice to the public is undertaken, Latin no longer serves the doctor in public health work and the simpler the language used by the engineer the more effective will his propaganda be.

However, apart from all considerations as to whether your technical language needs to be understood by anybody outside the engineering profession, sufficient evidence is accumulating steadily to prove that the time factor, or the time lag, must receive more consideration in studying the heating problem in certain types of jobs that are increasing in number month by month.

Before dealing with specific cases, let us imagine we are present at one of the preliminary meetings between a client and his architect and consulting engineer.

The architect describes the kind of walls he proposes to use in the building; the client, who has been reading some insulation advertisements, asks if that will be an insulated wall; the engineer replies that the transmission coefficient will be 0.278 Btu per hour.

The architect looks wise and is relieved when the client says, "What the 'Sam Hill' does that mean?"

Engineer: "That means that 0.278 British Thermal Units will flow through a square foot of wall when there is 1 F difference in temperature between the two sides."

Client: "Oh! is that so, well I think we should put something back of that wall to give us an air space."

Architect: "Well, we could put some 2 in. hollow tile at the back and plaster it, what would that do?"

Engineer: "That would give us 0.210 Btu."

Client: "Could we get anything better than that?"

Architect: "Oh yes; we could put on some furring and then plaster on ½ in. insulating board."

Engineer: "That would give us 0.147 Btu."

Client: "Can we do any better than that?"

Engineer: "Well, if we used a good thick insulation, say 2 in. thick, we would get 0.095 Btu and by using even thicker insulation we could go down as low as 0.05 Btu."

Client: "What do you think of it, Mr. Architect?"

Architect: "Oh, I don't know, that 0.210 looks like a pretty fair reduction to me and I've used that kind of wall before, so I guess we had better let it go at that."

Client: "Well, whatever you say, boys, you know better than I do."

Suppose the engineer had told them that the difference between these five walls was that number one would pass 1 Btu per square foot, when there was a difference in temperature inside to outside of about 3.6 F, number two the same amount of heat, when there was 4.76 F, number three 6.8 F, number four 10.52 F, and number five 20 F, would either the architect or the client have been much wiser?

But if the engineer had told them that, granted that the same temperature was maintained in the building, the amount of heat that would be wasted through the first wall in about 3½ hours, would take 4¾ hours to get through the second wall, more than 6¾ hours through the third, 10½ hours through the fourth, and 20 hours through the wall with the extra thick insulation, would the com-

parison have been more vivid and would there have been more study given to the economics of the whole question?

These, of course, are hourly differences for one degree difference on the two sides of the wall. For parts of Canada having a mean temperature differ-

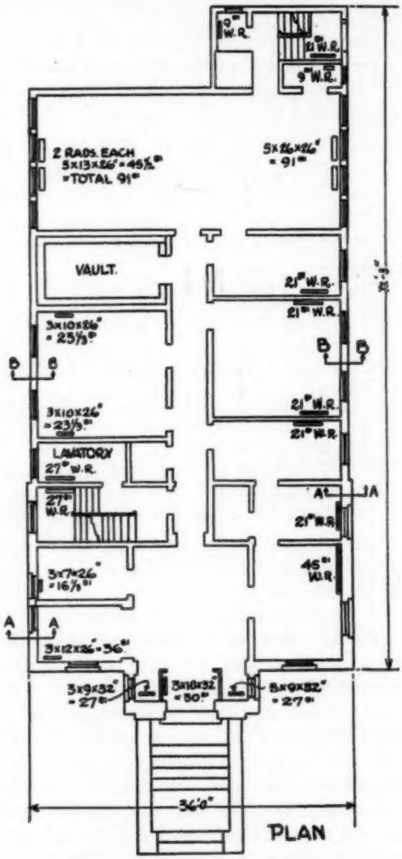


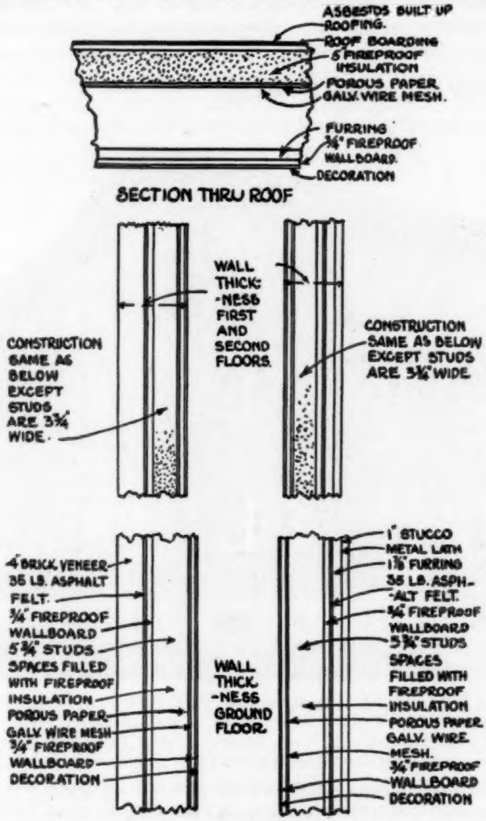
FIG. 1.—GROUND FLOOR PLAN

ence of 35 F between the inside of an average occupied building and the outside during a heating season of 210 days, these 5 Resistances could be expressed as 6.17 min, 8.16, 11.66, 18, and 34.28 min respectively.

But, inasmuch as the average person does not think of fuel in terms of a single Btu, the comparison might just as well be made in hours for enough Btu's to equal about 1 lb of coal—say, 12,000—and for an area of 100 sq ft of wall instead of 1 sq ft.

Then the figures would be 12.34 hours, 16.32, 23.32, 36, and 68.56 hours respectively, and thus we get a comparison based on values that mean something to any person of average intelligence.

Even with that explanation, our clients would probably thank us more to tell them that these differences, under the climatic conditions described, mean a dif-



WALL & ROOF SECTIONS

FIG. 1A.—DETAILS OF WALL AND ROOF SECTIONS

ference each season as between walls numbers one and five of approximately 2¾ tons of coal for each 1,000 sq ft of wall built.

While it is true that not many heated buildings, having a wall Resistance of 20 hours, have so far been built, several jobs have come under my own observation having that amount of resistance in roofs, and I have had personal experience with several jobs having both wall and roof resistance of 15 hours.

Such a resistance would give a saving of about 2½ tons of coal per 1,000 sq ft of wall built as compared with wall number one under the conditions outlined.

Changing the understanding of Resistance to mean time instead of degrees difference in temperature does, I am satisfied, create a much more definite mental picture for a man, who can imagine himself going down to the furnace and putting a match to so many dollars worth of fuel. He can easily be made to realize that with a certain type of construction his dollar's worth will last him, say, from 4 to 5 hours, and 15 or 20 hours with a different type.

The Effect of Heat Retention in Practice

So much then, for clearing up the misconceptions that arise through the use

FIG. 2—SMALL OFFICE BUILDING, WHOSE GROUND FLOOR PLAN IS SHOWN IN FIG. 1

of such terms as conductivity, coefficients of transmission, heat head or temperature difference, etc., etc., except when you are actually working at your desk or table or are gathered together at meetings of your society, the next point to be considered is what change, if any, does or will such high resistance make in heating engineering practice? Far be it from me to attempt to answer such a question. Your attention, however, is drawn to certain conditions that have been noted in some buildings and to the following considerations.

THE GUIDE recommends that the outside temperature to be assumed in the design of any heating system must not be more than 15 F above the lowest recorded temperature as reported by the U. S. Weather Bureau during the preceding 10 years for the locality in which the heating system is to be installed.

THE GUIDE further stipulates that as the lowest temperatures recorded are

usually of short duration, these conditions can readily be taken care of by the heat capacity of the building itself. These recommendations must have been based on actual heating experience in buildings that have proved satisfactory to occupants over a considerable number of years.

Inasmuch, therefore, as up till very recently, there were few heated buildings that had anything more than a layer of very thin insulation, the Resistance of the walls and roof of the average building would not probably exceed 5 hours.

If that resistance has proved sufficient to take care of the assumptions your Guide recommends, does it not follow that when the resistance is doubled and in some cases quadrupled, your assumptions should be modified?

From personal observation of a number of jobs during the past few years, I think it is fair to say that many members of your Society having good professional reputations do not assume outside temperatures 15 F above the lowest recorded for a 10 year period, but prefer a more conservative figure from 5 to 10 deg lower than the 15 deg margin would give.

Some of your Toronto members, for instance, have assured me that, theoretically, they could figure their work down to zero, but in actual practice they find that jobs so estimated are liable to give trouble several days in each season.

Other so-called heating engineers with no professional reputation to lose are not so reluctant to take a chance and do figure down to about zero. When they deal with jobs having wall and roof resistances of 5 hours or even less, the results are generally bad whenever the outside temperature drops very much below zero.

My contention is that the development of construction having resistances of 10, 15 or 20 hours makes it possible for the most conservative and reliable engineer to assume that the heat retaining property of such buildings will take care of all possible contingencies below the point generally figured by his competitors. On the other hand, the irresponsible cut price individual—in Toronto for instance—would have difficulty in persuading his customers to let him figure to about 10 deg above zero, because temperatures of 14 to 16 deg below zero are recorded—unofficially—quite frequently in Toronto and neighborhood.

It is unfortunate that, at the time of writing this paper, definite scientific data on support of my contention are not available in the form you are used to in the presentation of papers for your meetings.

I do submit, however, the ground floor plan and details of wall and roof sections, Figs. 1 and 1A and Fig. 2, showing a small office building in which I have had an opportunity of noting, pretty carefully, the operating results during the past winter.

This building stands exposed on all four sides, has no storm sash or storm doors, single thickness Vita-glass in windows, weatherstripped. The sashes are of the superior type each sliding vertically, while it is also possible to pivot them from the bottom so that they open inwards at the top and thus provide ventilation without direct draft on anyone sitting close by a window.

I mention this detail because it has a very decided bearing on the heating results obtained in the building; first, because the type of window encourages opening for ventilation, and second, because, although I have described it as weatherstripped, it has features that make it impossible to compare it with an ordinary double hung window weatherstripped.

The ground floor wall resistance was figured as 18.86 hours, transmission coefficient 0.053; the walls of two upper stories 14 hours, coefficient 0.071;

roof 15.15 hours, coefficient 0.066. Heating was calculated to zero and 160 Btu were assumed per square foot of radiation.

A gravity magazine feed type boiler is used and the fuel buckwheat anthracite. The boiler is not yet covered and mains in basement have only been covered since about the end of February; in the latter case the resistance provided is at least twice that usually given in ordinary commercial thickness coverings.

The results so far observed are that on the coldest day noted during the winter, when 22 F below zero was recorded in the morning, the temperature of the water at the boiler about 10 A. M. was 124 F and during the forenoon a tour of the building showed that a large number of the windows were open at the midrail for ventilation purposes.

In one room on the top floor, where I spent most of the day, the radiator was shut off entirely about 10 A. M. and was not opened again until 6 P. M., and during that time the window was slightly open at the center.

The caretaker advises me that since he has assumed control of the plant, the highest he has noted the temperature at the boiler was 135 F and that rise has only occurred when wind conditions cause excessive draft in the chimney and he cannot keep the temperature of the water at the boiler lower, as would be desirable.

My own examinations of the thermometer at the boiler have shown fluctuations from 98 F to 124 F and the building as completed is comfortable throughout every room.

Coal consumption for December, January, February and March averaged approximately 1 ton buckwheat per week.

Showing how results in this building compare with what is evidently accepted practice, the temperature of water, recommended by the manufacturers of the pressure control tanks used, in relation to outside temperature is as follows:

Outside Temperature	*Boiler Temperature*
60 deg above	120 deg
50 deg above	130 deg
40 deg above	145 deg
30 deg above	160 deg
20 deg above	170 deg
10 deg above	180 deg
0 Zero	190 deg
10 deg below	200 deg
20 deg below	210 deg
30 deg below	225 deg

Even though they add that these temperatures will vary according to the condition or construction of the building, it is obvious that the departure from these figures in actual experience need not have been so great if less radiation had been installed, which means that the job could have been figured for an outside temperature quite a bit above zero, or considerably more than 15 deg higher than the lowest recorded temperature for a 10-year period.

In case there should be any question as to whether the method of figuring the radiation for this building showed any departure from Society recommendations, the assumptions and calculations as actually used to determine the radiation for different rooms were submitted to one of the members of your Council.

Without checking all the calculations, but after examining my figures in some detail, he has expressed the opinion that the amount of radiation is not overesti-

mated, and is in accordance with the rules laid down for your members in the 1929 Guide.

Air infiltration was not figured by the crack method, because, although the sashes are weatherstripped, the particular type of sash and frame construction used is so different from any type on which I could find any record of tests in published documents.

With windows having two vertical cracks at the side of each sash instead of one, and only one of them weatherstripped, I did not feel that test figures could be made to apply. Estimates of air change were therefore made, and the changes assumed strictly adhere to Table 14, page 49, of The Guide 1929.

While I cannot give the same definite information about the Orillia Hospital Job, I can vouch for the fact that the best data, available at the time of con-

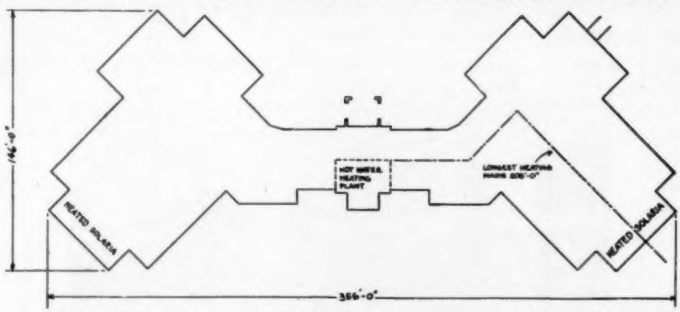


Fig. 3.—Ground Floor Plan of the Ontario Government's Hospital at Orillia

struction, were used, and the results are, as you will note, very similar to what the office building gives.

Forced Hot Water Heating

For forced hot water heating, page 117 of The Guide 1929 shows a chart of maximum water temperatures for different outdoor temperatures.

Contrast these figures, showing 130 F water at 50 F outside, increasing to 230 F water at 20 F below zero outside, with the actual experience on a forced hot water job at the Ontario Government's Hospital, Orillia.

There, as the result of wall and roof resistances of at least 12 and 20 hours respectively, I am advised by the operating engineer that, at about 30 F below zero outside, the maximum temperature of the water at the heater has not exceeded 150 F and that a maximum of 140 F will take care of the ordinary very cold dips that have been experienced during the four years of the operation of this part of their heating system.

Notwithstanding the wide-spreading area of this one-story hospital building, it has been found that for most weather conditions water can be circulated by gravity with a temperature drop of about 20 deg between the flow and return at the heater, and that the pumps are only required in very extreme low dips, when the flow and return temperature difference only varies from 4 to 8 deg.

As is the case with the office building previously referred to, the operator's chief difficulty is to keep the temperature of the building down with the plant as

installed, even although at the hospital many hundreds of feet of radiation, originally included in the installation, have been removed and utilized in other buildings. The tendency to open the windows in cold weather, noted at the office building, is also observable at Orillia.

Fig. 3 gives an idea of the area covered by the Orillia building.

Going further north to the still colder climate at the Paymaster Mine at South Porcupine, Ontario, one can find additional evidence of the need for revising heating assumptions as construction methods change.

The developments there are very fully dealt with in three articles in the *Engineering and Mining Journal* (New York) February 16, March 2 and March 16, 1929, *Economical and Efficient Housing in Hot and Cold Climates* by H. E. Clement and the writer.

Residence Heating

Fig. 4 shows two houses marked 1 and 2 and the following data are supplied

FIG. 4.—HOUSES NOS. 1 AND 2

by one of the largest firms of general contractors in northern Canada (dated April 6, 1929).

House No. 1: 26 ft x 33 ft; 12,500 cu ft contents in house proper and 5,900 cu ft in basement; coal consumed since last fall 7 tons; house heated evenly, having a uniform temperature of 70 F or slightly over at all times; 350 sq ft hot water radiation; boiler is a jacketed, hot water, domestic heater having a grate 15 in. in diameter, but not rated in radiator capacity as it is not ordinarily used for heating buildings; temperature of water 140 F with outside temperature from 25 to 35 F below zero; average temperature of water for three cold winter months 130 F. Occupant of this house is an engineer and is the master mechanic for the construction company. He reports that as the extreme cold weather only lasts from 24 to 48 hours, he could not see any very marked difference in the firing to keep the house at the same temperature during extreme cold weather. Resistance of walls and roof of house No. 1, approximately from 12 to 14 hours.

House No. 2: 25 ft x 29 ft, much less cubic contents than house No. 1; has 450 sq ft hot water radiation, average temperature water during three severe winter months 160 F; house not nearly so warm or so evenly heated and consumes more coal. House No. 2 was built by same construction company and was of much better construction than ordinary frame buildings as the sheathing outside of studs is double with felt between; studs sheathed with lumber inside,

then felt and lath and plaster; sunroom in House No. 2 not heated, whereas in No. 1 it is built in with house and heated.

Skyscraper Possibilities

Up to the present the results I have described have not been economically possible in skyscraper construction, but I am satisfied that within the next few years materials will be developed that will make possible wall and roof resistances of 15 to 20 hours.

The greatest stumbling block to such progress is the difficulty in getting the building by-laws in our large cities changed to allow engineers and architects to take advantage of the results of modern research work.

The demand for increased height and consequently reduced weight will compel the use of materials having very much greater resistance to heat flow than those now commonly used. Such materials are available now and all that is needed is to adapt them to skyscraper requirements.

I have the figures for a recently erected 16-story building in Toronto before me, and I note that the resistance of the wall was figured as 3 1/3 hours and that nearly 5,000 sq ft of radiation was required for wall losses. With this resistance increased to 20 hours, as will soon be possible, 800 sq ft of radiation would take care of the wall losses, a reduction of 4,200 sq ft. As the total radiation for the building was 10,000 sq ft, the saving would represent at least 40 per cent and as it would not be necessary to figure down to as low an outside temperature the saving would probably be even more than 40 per cent. In dollars and cents at present prices in Toronto, it would amount to at least $7,000.00 and for this particular building that represents about 15 cents per sq ft of exterior wall.

The new construction materials to give such results will not be added to the existing types, but will take the place of some of them. Even with present developments known to research organizations, an allowance of 15 cents per sq ft of wall or roof would go a long way towards making up any difference between the cost of what we now do and what we will do in the near future.

Intermittent Heating

In buildings where the heating problem is complicated because of intermittent use, such as churches, club-rooms, etc., a substantial increase in resistance to heat loss offers a much more economical solution of the problem than the provision of excess stand-by heating equipment.

My attention has been drawn to several cases, where the amount of heating plant installed for this purpose represents not only considerable capital outlay, but also a very wasteful annual expenditure for maintenance and operation.

In this connection, it should be noted that while the work already done shows that high resistance will permit large reductions in the amount of radiation, care should be taken in cutting down the sizes of heating mains and boilers. Free and rapid circulation is essential.

These jobs I have had to do with personally have been successful, I believe, because my associates have given special attention to this feature of the work.

Use of Gas and Electricity for Heating

Your members know more than I do about the future possibilities in the use

of gas, electricity, oil, etc., for heating all kinds of structures, but I think you will agree that their fullest development will only be possible when we, as architects, provide you, as engineers, with wall and roof resistance sufficient to bring the cost to the consumer within the reach of a much larger clientele than will now consider such methods. However, the demand for labor saving equipment is spreading and your practice along such lines will undoubtedly be influenced by your co-operation in educating the public, and educating the public can only be done by using language it can understand.

Conclusion

I may not have submitted enough technical data to show that the points raised in this paper are of any serious interest to practicing engineers.

I submit, however, that no Society, professional or otherwise, can achieve its greatest end, namely, service to its members, when these members are dependent on the support of all the people, unless it makes its work known to the greatest possible number of people.

This paper, then, is a plea for a wider diffusion of knowledge of heating possibilities by using language that would make such diffusion possible.

DISCUSSION

PROF. L. M. ARKLEY: In regard to Mr. Govan's paper, the first suggestion that he makes is in connection with our definition of resistance, and I think perhaps that it is a good one, because when you tell a layman that your conductivity coefficient is based on the Btu transferred per square foot, per degree difference in temperature, per hour, he is all up in the air. So it seems to me Mr. Govan's suggestion might be well taken.

In regard to the second feature of the paper where he suggests basing our computations on an 18- or 20-hour resistance factor, I think it will perhaps be some time in the future before we can do that, because there are no buildings or very few that are built on that assumption at the present time.

S. R. LEWIS: I have had some experience with warehouses along the line Mr. Govan suggests, and I have had the same rather interesting conclusions. When we are talking about putting in a heating plant, we do not care very much about the efficiency of the building construction. We will burn more coal, and do not worry about it. When we have to do with other things to be heated, lo and behold, we are subconsciously building the building so well that we do not need any heating plant.

R. C. BOLSINGER: I would like to ask the speaker whether he found the same conditions existed when they used steam for heating the buildings as they did when they used water.

JAMES GOVAN: I am sorry to say, I have no experience at all with steam in a building having a high resistance such as I have spoken of today. It so happens that all of the buildings into which resistance of that kind has been built that have come to my attention, have been heated with hot water, and indeed, it is fortunate that that has been so, because if we had ordinarily steam heated buildings with the kind of resistance built in there, you can see what would have happened. If we circulate water at 130, when ordinary calculations show we should be doing it at 200, what would happen if we had steam where we

could not modify the temperatures in the radiators, except we had been using some sort of modification system of steam. I have had no experience to answer the last question.

C. L. RILEY: I think the tendency that we are now finding in some of our work to introduce cool air in the summer, will force upon us the consideration of such factors as Mr. Govan has spoken of. Heretofore, we have simply thought of the heat needed in winter in a building, and have taken the building as we found it, or as the architect designed it, and we have not concerned ourselves with the construction of the building, except in a general way perhaps, and we have put in a heating plant which will provide heat enough to heat the building as it is. Now, our job is going to be in the very near future, when we get into refrigeration in summer, to take in hand the entire design and provide buildings which will hold the cold in summer and the heat in winter, along the lines that are suggested by Mr. Govan. I think it is right ahead of us just as he says, but we have not had such severe conditions as he has had to deal with, so it is very valuable for us, in view of what we shall soon have to do, to have some of these figures on record.

G. T. PEARCE: I think Mr. Govan's suggestion in regard to putting the question simply to the people who are to use your materials is very excellent. We in the refrigeration industry have a condition that is rather difficult to combat. We have had a condition for a period of years—whether it is due to the fact that those who talked about insulation for refrigeration talked in terms which were not intelligible to the general public, or whether it was due to the fact that they wanted to have others feel that they knew more about the subject than anyone else could—but the general public when you talk to them about insulation, figures that you have to have a certain definite ability, perhaps mysterious or otherwise, and unless you make those particular passes at the time that you are building a building, or building a structure, your result is not going to be right.

Now, any boy who has finished the eighth grade can take the information that is available and figure out what the heat transmission is through that building structure, or any kind of an edifice, and do it without going beyond the studies that he has achieved up to the eighth grade. Thus, we take the mystery out of the proposition, because we are having a very hard time to get the mystery out of refrigeration. Get it out in the beginning instead of waiting until later!

PROF. F. E. GIESECKE: I consider this paper a very valuable contribution to our literature. It has given me a new idea and, I believe, a very valuable one.

MR. LEWIS: If I am not mistaken, during the war, statistics showed that we burned about as much coal per square foot of installed radiation in Texas as we did in Northern Minnesota. As you go north we build buildings better, and so use less fuel proportionately.

L. A. HARDING: I should like to ask Mr. Govan how the infiltration was calculated in these buildings—on the basis of leakage of periphery of the sash or on air changes—THE GUIDE gives both and it sometimes makes quite a difference in the results obtained in the building.

MR. GOVAN: That question is answered in the paper itself, but I might explain that the particular window used in this building was one that not only slid up and down, but the sash was divided all around in such a way so as to

get double cracks. It is a sort of compound sash that opens as a casement. That means that while we weatherstripped the sliding portion, there was another portion there that we did not weatherstrip, and I could not find anything in THE GUIDE that would give me any information at all that could be used with regard to those particular windows. We took the information in THE GUIDE as to air changes, therefore. In other words, the exposures, whether the windows are on two sides, etc., were noted, and the instructions in THE GUIDE as to air infiltration were very carefully followed.

Anticipating a question of that kind, I asked a member of your Council, H. H. Angus, to go over the figures for one or two of the rooms in that particular office building with me, to see whether the calculations were in excess of what your GUIDE called for. He assured me that not only were they not over-estimated, but probably if anything, were under-estimated.

On the other hand, as I said in my delivery, on the coldest day in the winter, three of us went around that building and found that 50 per cent of the windows in all the occupied rooms were open when the outside temperature during that forenoon was 22 below zero. That is the answer to the question as to whether the infiltration really is at the bottom of the differences that we are getting from THE GUIDE figures, and the same thing applies to the hospital building in Orillia, Ontario. They are having difficulty keeping the windows closed. It is so warm even at 30 below zero, that they will open windows, and the administrators of the building have told me repeatedly that they wish they could find some way of controlling the windows being left open at night.

CHAIRMAN LEWIS: Mr. Govan, if I might introduce a remark here—you know the Scotch are a hardy race.

MR. GOVAN: Well, that hospital building, Mr. Lewis, is a hospital for feeble-minded, and it is not occupied by the Scotch.

MR. HARDING: Do you attribute this difference to apparently mistaken values that we are using for overall transmissions in THE GUIDE? Do you think that is where your trouble lies?

MR. GOVAN: No, I think that your GUIDE figures are quite all right for that, but the assumption that you make—take our 20 year period, the lowest temperature for 20 years or so, and then go 15 deg above that, does not apply to a well insulated building. In other words, the heat retained over a period of 10, 12 and 14 hours, will see you over your low depths, and you can very well come away up on buildings such as I am describing. Instead of figuring down to 10 below, you could in Toronto, for instance, figure at 10 above, and it is that difference that is playing hobb with all your GUIDE recommendations.

MR. HARDING: What do you figure restores this heat in the building, the air or furniture or what?

MR. GOVAN: No, the fact is that the heat cannot get through the wall. In other words, as I said, if the wall and roof construction will hold a given amount of heat under a given temperature condition for 20 hours, whereas the ordinary old type of walls only held it for 4 or 5 hours, that is why you take care of the low depths, because the low depths usually do not last for more than a few hours.

IN MEMORIAM

NAMES	JOINED THE SOCIETY	DIED
HENRY ADAMS*	Charter Member	Dec. 1929
EMERIT E. BAKER	1910	Jan. 1929
WILLIAM D. CLARK	1908	Jan. 1929
RALPH A. FLEMING	1926	May 1929
ELLSWORTH H. FRITZ	1929	Nov. 1929
JOHN GORMLY*	Charter Member	Jan. 1929
EDMUND GRASSLER	1919	Oct. 1929
EDWIN S. HALLETT	1918	Mar. 1929
AUGUST KASTELLO	1923	Nov. 1929
IRVING LONDON	1924	July 1929
LUTHER B. MCMILLAN	1918	Aug. 1929
HERBERT MUTH	1912	Dec. 1929
DAVID M. NESBIT	1895	Mar. 1929
JAMES C. NORRIS	1928	Jan. 1929
LEONARD G. PAINE	1920	Apr. 1929
HENRY W. WENDT	1917	June 1929
MORRIS G. WHITE, JR.	1925	Jan. 1929

*Past President

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